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STUDY OF A HEAT REJECTION SYSTEM USING CAPILLARY PUMPING

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Prepared by
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16. Abstract

Results of an analytical study investigating the application of capillary pumping to the heat rejection loop of an advanced Rankine Cycle power conversion system are presented. The feasibility of the concept of capillary pumping as an alternate to electromagnetic pumping is analytically demonstrated. Capillary pumping is shown to provide a potential for weight and electrical power savings, and in increasing reliability through the use of redundant systems. A screen wick pump design with arterial feed lines was analytically developed. Advantages of this design are high thermodynamic and hydrodynamic efficiency, which provide a lightweight system, easily packaged. Operational problems were identified which must be solved for successful application of capillary pumping. The most important are development of start-up and shutdown procedures, of a means of keeping non-condensibles from the system, and of earth bound testing procedures.

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FOREWORD

The research described in this report was conducted by the TRW Systems Group under NASA contract NAS 3-12976. Mr. James P. Couch of the Lewis Research Center Space Power Systems Division was the NASA Project Manager.

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SUMMARY

The Advanced Rankine Cycle Space Nuclear Power System of current interest to NASA/Lewis Research Center is a three-loop configuration in which the primary loop transfers heat from a nuclear reactor to a power conversion loop where electric power is generated. Waste heat from the power conversion loop is transmitted by a third loop to a space radiator for ultimate rejection to space. The heat rejection loop (actually four parallel units) employs electromagnetic pumps for transport of the coolant fluid. The analytical study documented in this report examined the feasibility of using passive capillary pumping in the heat rejection loops.

The capillary pump concept is a two-phase system which transports thermal energy as latent heat of vaporization. The motive force for circulation of the working fluid is the pressure head generated by a capillary structure in the heated region. The capillary pumped system, contrary to the conventional heat pipe, flows vapor and condensate in the same direction in a closed system; separation of the liquid and vapor occurs in the condenser where the liquid bridges the flow passage. Also, the conventional heat pipe has wicking throughout the system whereas the capillary pumped loop can be wicked only in the heated zone.

The study documented in this report consisted of four major activities:

A literature survey to obtain information about capillary pumped systems, condensing heat transfer and thermodynamic property data for the candidate working fluids (sodium, potassium and cesium), and operating characteristics of Rankine cycle systems in space.

Analytical representation of a number of pumper conceptual designs and wick configurations. A design computer code was formulated to permit optimization of the heat rejection system (pumper, radiator, lines) for the imposed heat loads.

Formulation and operation of a performance computer program to evaluate the characteristics of many coupled heat rejection loops for design and off-design operating conditions. The effect of one or more radiator failures was determined.

Identification and examination of operating difficulties associated with liquid metal capillary pumped systems. Where possible, design solutions to these problems are suggested.

The wick/condenser configurations evaluated included the Stenger design where liquid is introduced at the center and heat is supplied to the exterior of a porous plug wick, a conventional wick design which uses arterial headers to supply liquid, and the screen wick concept where the conventional wick is replaced by a simple screen displaced slightly from the heated wall. In the latter case, heat transfer is through the liquid space which also serves to distribute liquid from the arterial feeds to the screen surface. The analysis showed that the screen concept is substantially smaller and lighter than the other two concepts. The heat transfer area requried (80 ft^2) was comparable to that required by the EM-pumped configuration (60 ft^2).

The results showed that potassium is the preferable working fluid. Sodium was eliminated because its vapor pressure (@ 950°F) is not sufficient to maintain flow. Cesium was found inferior to potassium in its chemical properties, its capillary pumping capability, and its

thermal conductance. Further, the potassium system is slightly lighter. It was not possible to establish the optimum number of heat rejection loops with finality. Comparison of the system weights showed a broad minimum between about 12 and 24 loops, but the weight variation between the minimum and maximum was only 10%. From a control point of view, it appeared that about 24 or more loops would be desirable. Failure of one loop would then represent only a small perturbation to the total system. The reliability, maintainability, and manufacturability are important variables in making this decision. The preliminary design was based on a twelve loop system, although the performance characteristics can be easily related to the 24-loop configuration.

Operational problems of capillary pumped systems have been identified by this study and by others. Those problems which appear to be most troublesome are start-up and shutdown of the system without freezing in the radiator; the possibility of non-condensible gases accumulating in the system resulting in interruption of the pumping, and operation in the presence of acceleration forces, both on the ground and in space. Potential solutions for these problems, including a gas trap for collection of non-condensible gases and a wicked reservoir to collect liquid during shutdown operations are briefly described. It is difficult to analytically evaluate the effectiveness of these concepts.

It was concluded that the capillary pump heat rejection system concept is feasible as a replacement for conventional EM pumped systems and potentially offers substantial advantages in weight, electrical power and reliability. A number of troublesome operational difficulties associated with a capillary pumped concept remain to be resolved.

1.0 INTRODUCTION

The advanced Rankine-cycle space-electric-power system has been under study for several years by the National Aeronautics and Space Administration. The configuration of current interest is a three loop system in which the primary loop transfers heat from a nuclear reactor to a power conversion loop where electric power is generated. Waste heat from the power conversion loop is transmitted by a third loop to a space radiator for ultimate rejection to space. The heat rejection loop (actually four parallel units) employs a counterflow condenser, an electromagnetic (EM) pump and a space radiator. These components represent a sizeable fraction of the overall system weight (approximately 18% of the total for an unmanned spacecraft version) and require a total of 11 KWe to drive the pumps. Of perhaps more concern is the reliability of the present heat rejection loops for the long mission times considered and the severe weight penalties associated with providing system redundancy.

The study documented in this report examined the feasibility of replacing the present EM-pumped heat rejection loops with a completely passive capillary pumped system. This configuration, first proposed by Stenger (1)*, is a two-phase system which transports thermal energy as latent heat of vaporization rather than sensible heat of a liquid. The device, shown schematically in Figure 3-1, uses the capillary pressure head generated by the wick in the heated area to provide the motive force for circulation of vapor to the condensing radiator and for liquid flow back to the heat source. Capillary pumping is a passive surface tension phenomenon with no electrical or mechanical energy requirements and potentially avoids the power and reliability limitations of EM-pumping.

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* Underlined numbers in parentheses refer to cited References listed following Section 7. Conclusions.

A survey of the literature showed that studies of capillary-pumped systems, comparable to the one studied here, have been extremely sparse. References 1, 2, and 3 are the only contributions which could be found. For the most part, the work reported in these references was experimental in nature and provided very little analytical background. The work discussed in this report is the first major step to analytically characterize the performance of the capillary-pump system for a particular application and to examine the operational advantages and difficulties encountered in that application.

The primary objective of the present study was to determine the feasibility of using capillary pumping in the advanced Rankine cycle power system heat rejection loops. In support of this goal, the following tasks were performed:

- A literature survey was conducted directed toward obtaining information about capillary pumped systems, condensing heat transfer and thermodynamic property data for the candidate working fluids (sodium, potassium, and cesium), and operating characteristics and difficulties of Rankine cycle systems.

- An analysis of various conceptual pump designs and wick configurations were made to establish a design of minimum size. A "design" computer program was formulated to optimize system size (minimum weight) for the imposed heat loads.

- A "performance" computer program was formulated to compute the performance of a series of capillary pump heat rejection loops. This program was also used to assess performance variations resulting from a radiator failure and other off-design conditions.

- An attempt was made to identify and resolve operating difficulties associated with capillary pump systems. For the most part, these operational problems are not amenable to analytical treatment and would require experimentation for their complete identification and resolution.

Section 4.0, Systems Analysis Model, describes the general heat rejection system configuration and presents the equations to simulate each of its components. Section 5.0, System Preliminary design, gives the results of applying the design and performance programs to determine the optimum working fluid, the number of heat rejection loops, the final geometric configuration, and to calculate the performance of the complete heat rejection system. Section 6.0, Systems Operational Problems, gives a discussion of operational problems, and potential solutions to these problems.

The Appendices contain the details of the analytical work. Appendix A, Conceptual Design Analysis, gives the mathematical analysis of specific capillary pump configurations. Appendix B, Fluid Property Equations, gives the capillary pump working fluid (sodium, potassium, and cesium) property equations obtained from the literature. Appendix C, Design Computer Program; Appendix D, Performance Computer Program; Appendix E, Radiator Optimization Program; and Appendix F, Radiator Performance Subroutine, RADP; each gives a complete description of the computer programs used in this study. The description is in sufficient detail that the implementation of these programs on a digital computer can be accomplished quickly by any interested organization. Finally, Appendix G, Nomenclature, summarizes the mathematical symbols used in the body of the report.

2.0 GENERAL SYSTEMS REQUIREMENTS

This study of capillary pumped heat rejection systems was directed specifically to application for the Advanced Rankine Cycle Power System (ARCPS). The operating conditions of the ARCPS and the environment of its proposed space application, therefore, established the general systems requirements and constraints on the program reported. This section provides a summary of these requirements as well as a brief description of the operation of the ARCPS. This information was extracted from the results of design studies made by NASA/Lewis Research Center; the reader is referred to Reference 4 for a more complete account of the power system.

The ARCPS is a space power system employing a three loop liquid metal circuit as shown schematically in Figure 2-1. A nuclear reactor in the primary loop is cooled by liquid lithium. The lithium, circulated by an EM pump, passes from the reactor and through a counter flow boiler where heat is transferred to the potassium working fluid of the secondary loop. Superheated potassium vapor from the counterflow boiler passes to a turboalternator, a condenser, an EM pump, a preheater and back to the boiler. In the turboalternator the thermal energy is transformed into electrical energy so that the exit potassium stream is of low pressure and subsaturated temperature. Moisture separators in the turboalternator divert hot liquid to furnish heat for the boiler preheater. The potassium vapor is condensed by the heat rejection loop in a counterflow condenser. In the conventional system, the working fluid (NaK) is pumped from the condenser to radiators for waste heat rejection to space.

Multiple loop heat rejection systems permit failure of a single loop without total loss of the system. As presently conceived, there are four separate heat rejection loops, each with its own condenser, pump and radiator. Approximately 75% of system capacity can be maintained even if one loop fails; e.g., by a meteoroid penetration. The number of loops has been selected so that the system meets this end-of-life requirement.

NET THERMAL INPUT POWER	2.0 MW
NET SHAFT POWER	433 KW
ALTERNATOR OUTPUT	398 KW_e
NET UNCONDITIONED POWER TO USER	345 KW_e
NET SYSTEM EFFICIENCY	0.172

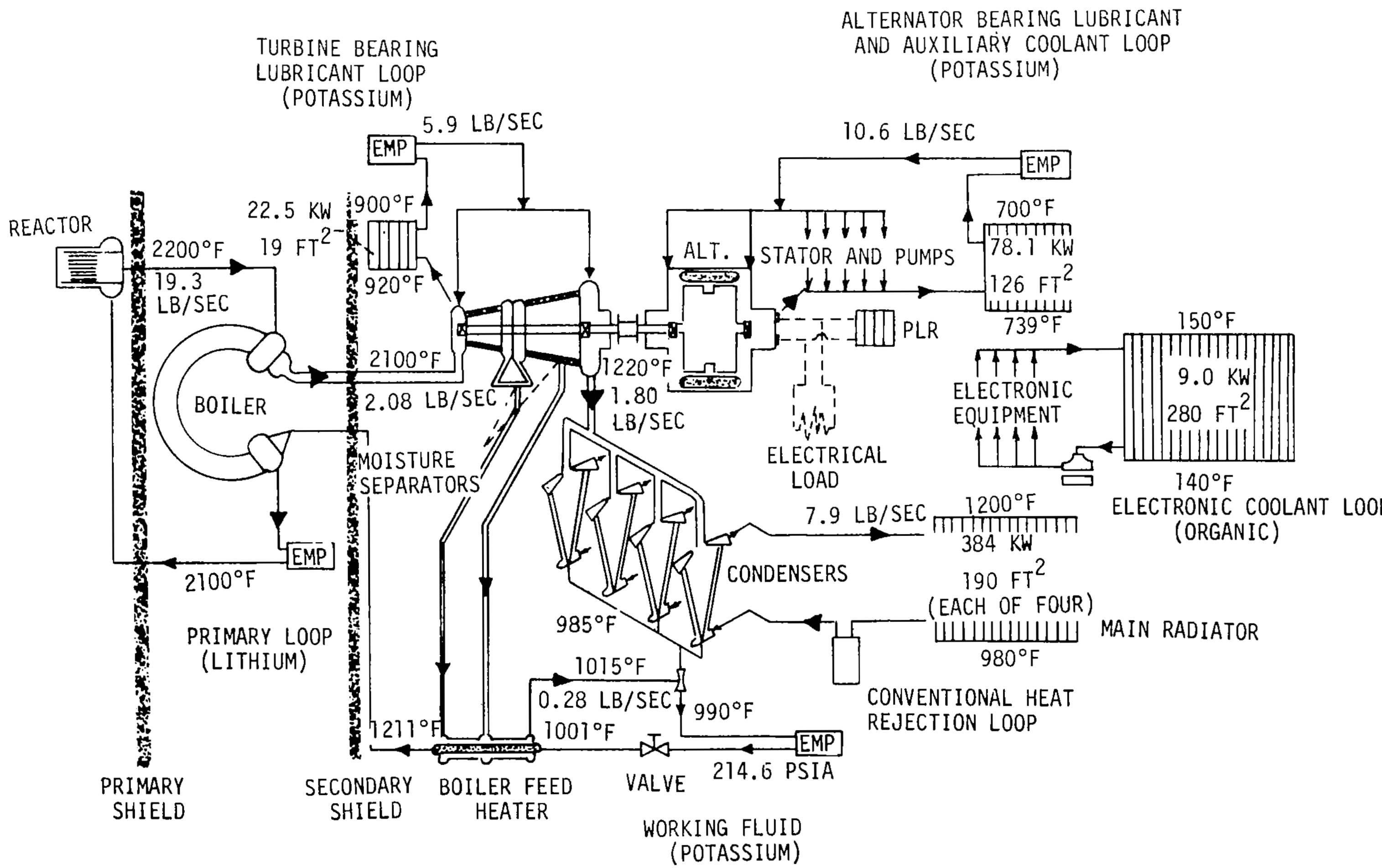


Figure 2-1. ADVANCED RANKINE POWER SYSTEM

NOMINAL 300 KW_e

Table 2-1 presents pertinent system parameters for the ARCPS with four conventional heat rejection loops; these data resulted from design calculations provided by NASA/Lewis Research Center. For comparison purposes, Table 2-2 provides a weight summary of the conventional ARCPS. Specific characteristics of the EM pumps and space radiators for the conventional heat rejection loops are described in Table 2-3. Additional general systems requirements imposed on this study by NASA/Lewis Research Center are listed in Table 2-4.

The geometrical constraints placed upon the system are illustrated in Figure 2-2. The reactor is to be placed at the cone apex with a shadow shield at its base. These components are located above A-A in Figure 2-2. The potassium boiler, pump, turbine, and turboalternator are located in the section just below A-A. The capillary pump is to be placed at the top of the following section, as shown in Figure 2-3. The main radiators, to which the pump rejects heat, are to be located around the outside of this latter section.

TABLE 2-1. NOMINAL 300 kWe ADVANCED RANKINE CYCLE POWER SYSTEM

I. OVERALL CYCLE

Reactor power (net to system)	kWt	2000	
Unconditioned power to user	kWe	345	
Cycle efficiency, net (uncond. to user/reactor)	%	17.2	

II. PRIMARY LOOP (LITHIUM)

Reactor outlet temperature	°F (°K)	2000	(1478)
Reactor temperature drop, ΔT	°F (°K)	100	(56)
Pump inlet pressure	psia (N/m^2)	40	(275800)
Loop pressure drop, ΔP	psi (N/m^2)	12	(82740)
Flow rate, lithium	#/sec(Kg/sec)	19.3	(8.75)
Pump input power	kWe	11.6	

III. POWER CONVERSION LOOP (POTASSIUM)

Boiler inlet temperature	°F (°K)	1211	(928)
Boiler inlet pressure	psia (N/m^2)	184.6	(1272820
Turbine inlet temperature	°F (°K)	2100	(1423)
Turbine inlet pressure	psia (N/m^2)	163	(1123890
Potassium superheat	°F (°K)	73	(41)
Interspool moisture separation temperature	°F (°K)	1520	(1100)
Turbine quality at interspool separator	%	90	
Interstage moisture removal temperature	°F (°K)	1300	(477)
Turbine quality at interstage removal stage	%	90	
Turbine exit quality (@η_t + 0.80)	%	89	
Condenser inlet temperature	°F (°K)	1220	(933)
Condenser inlet pressure	psia (N/m^2)	5.4	(37230)
Condenser exit temperature	°F (°K)	985	(803)
Condenser exit pressure	psia (N/m^2)	3.4	(23440)
Boiler feed pump inlet temperature	°F (°K)	990	(823)
Boiler feed pump pressure rise, ΔP	psi (N/m^2)	211.2	(1456220
Boiler preheater inlet temperature	°F (°K)	1001	(811)
Boiler preheater exit temperature	°F (°K)	1211	(927)
Boiler flow rate	#/sec(Kg/sec)	2.08	(.943)
Condenser flow rate	#/sec(Kg/sec)	1.80	(.816)
Feed Pump input power	kWe	10.7	

IV. MAIN HEAT REJECTION LOOP (NaK)

Radiator outlet temperature	°F (°K)	980	(800)
Radiator inlet temperature	°F (°K)	1200	(922)
Heat load, 4 segments, total	kWt	1536	
Flow rate, NaK, per segment	#/sec(Kg/sec)	7.9	(3.58)
Pump inlet pressure	psia (N/m^2)	10	(68950)
Pump pressure rise, P	psi (N/m^2)	12	(82740)
Radiator loss	psi (N/m^2)	4	(27580)
Condenser loss	psi (N/M^2)	5	(34475)
Piping loss	psi (N/m^2)	3	(20865)
Pump input power, per segment	kWe	2.75	
Radiator area, per segment, actual	sq-ft (m^2)	190	(17.7)

TABLE 2-1. NOMINAL 300 kWe ADVANCED RANKINE CYCLE POWER SYSTEM (Cont'd)

V. ALTERNATOR LUBE AND COOLANT LOOP (POTASSIUM)

Heat load, total	kWt	78.1	
Radiator outlet temperature	°F (°K)	700	(644)
Radiator inlet temperature	°F (°K)	739	(666)
Flow rate, K	#/sec(Kg/sec)	10.6	(4.81)
Pump pressure rise, ΔP	psi (N/m^2)	25	(172380)
Pump input power	kWe	7.3	
Radiator area, actual	sq-ft (m^2)	126	(11.7)

VI. TURBINE LUBE AND COOLANT LOOP (POTASSIUM)

Heat load	kWt	22.5	
Flow rate, K	#/sec(Kg/sec)	5.9	(2.68)
Pump pressure rise, ΔP	psi (N/m^2)	10	(68950)
Radiator exit temperature	°F (°K)	900	(755)
Radiator inlet temperature	°F (°K)	920	(766)
Pump input power	kWe	1.7	
Radiator area, actual	sq-ft (m^2)	19	(1.77)

VII. ELECTRONICS COOLANT LOOP (DC 200)

Radiator outlet temperature	°F (°K)	140	(333)
Radiator inlet temperature	°F (°K)	150	(339)
Heat load, total	kWt	10.4	
Flow rate, DC 200	#/sec(Kg/sec)	2.1	(.953)
Pump pressure rise, ΔP	psi (N/m^2)	60	(413700)
Pump input power (PMA)	kWe	1.2	
Radiator area, actual	sq-ft (m^2)	325	(30.2)

TABLE 2-2. NOMINAL 300 kWe ADVANCED RANKINE CYCLE POWER SYSTEM WEIGHT ESTIMATES

	Lbs	(Kg)
Reactor (2 MWt, 0.7 B.U., 20,000 hr-life)	1000	(499)
Reactor controls, drive, structure	200	(91)
Lithium primary loop, total	1500	(680)
Power conversion loop, K, total	2930	(1329)
Main heat rejection loop (NaK), total	3480	(1578)
Radiators panels (4), 2170		
Pumps (4), 480		
Expansion tanks (4), 400		
Piping, 430		
NaK inventory, included in above		
Alternator coolant loop, K, total	580	(263)
Turbine coolant loop, K, total	210	(95)
Electronic cooling loop, DC-200, total	530	(240)
Electrical equipment, total	1170	(531)
Structure (component & piping supports, radiator supports, etc.), assumed	3000	(1361)
Shielding shadow, unmanned, 10° halfcore angle, 150 ft. distance to payload), total		
Unmanned primary, 4000		
Manned (0.1% fission products leakage), 10,000 hrs.		
Primary, 3000		
Secondary, 7000		
Total: Unmanned	18,700 lbs=54.2 lbs/KWe net (8482 Kg=24.6Kg/KWe net)	
Manned	24,700 lbs=71.6 lbs/KWe net (11204 Kg=32.5 Kg/KWe net)	

TABLE 2-3. CHARACTERISTICS OF EM PUMPED HEAT REJECTION COMPONENTS - NOMINAL 300 kWe ARCPS

PUMP

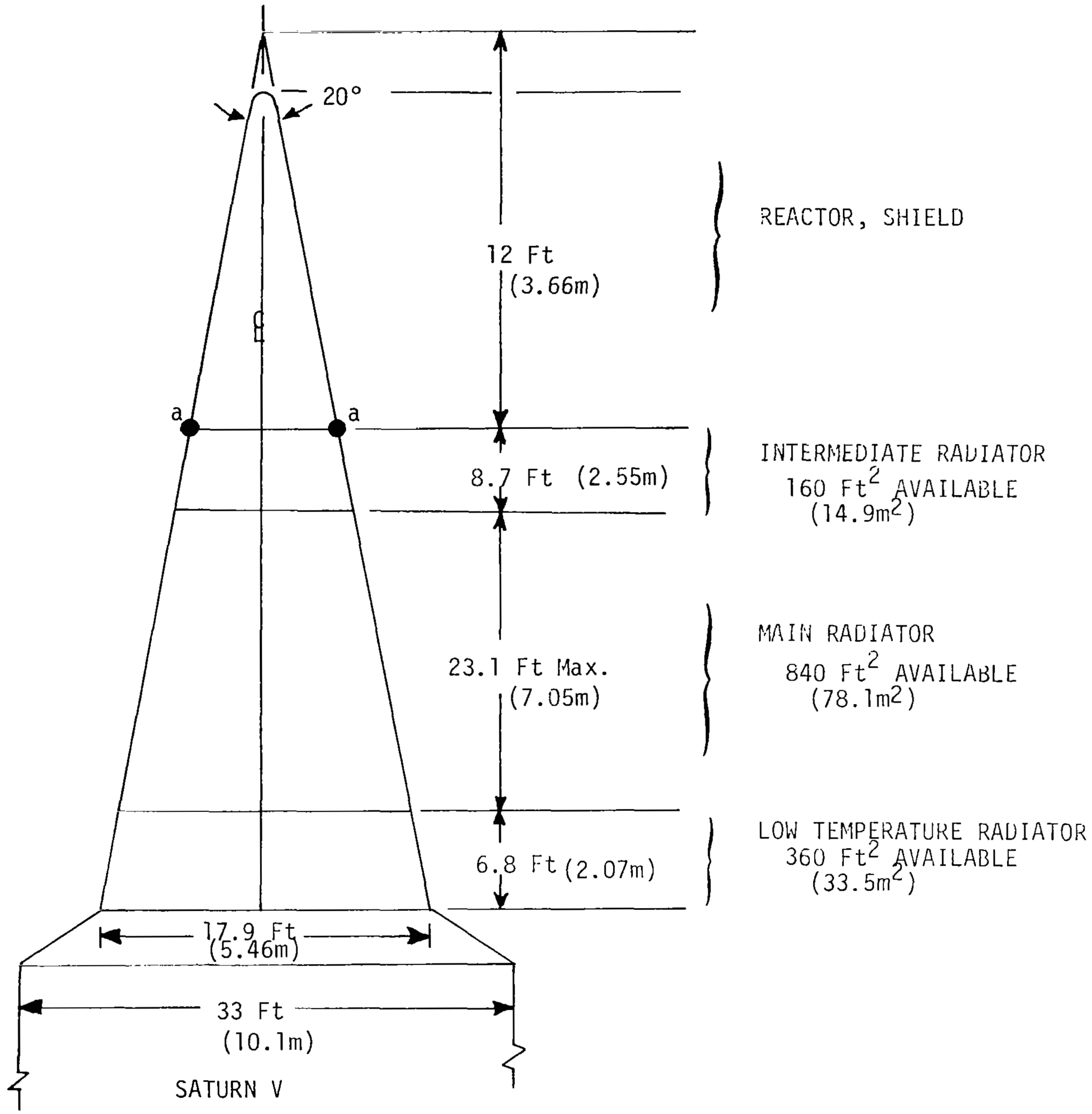
Type	EM
Number	4
Fluid	NaK
Flow rate, #/sec (Kg/sec)	7.9 (3.58)
Head rise, psi (N/m^2)	12 (82740)
Inlet pressure, psia (N/m^2)	10 (68950)
Input power, kWe	2.75
Efficiency, %	15
Inlet temp., °F (°K)	980 (800)

RADIATOR

Type	Armored tube and fin
Number	4
Fluid	NaK
Maximum Temp., °F (°K)	1200 (922)
Temp. drop, °F (°K)	220 (122)
Tube & armor material	SS
Fin material	SS clad Cu
Tube, i.d., in. (m)	0.17 (.00432)
Fin, thickness, in. (m)	0.025 (.00635)
Meteoroid protection	0.99 probability of no puncture in 20,000 hours
Survivability	3 out of 4
Fin effectiveness, %	93.6
Coatings	iron titanate
Coating emittance	0.90
Coating absorptivity	0.75
Effective sink temp., °F (°K)	105 (314)
Total area, sq-ft (m^2)	760 (70.6)

TABLE 2-4. GENERAL SYSTEMS REQUIREMENTS FOR CAPILLARY PUMPED HEAT-REJECTION SYSTEM

Heat rejected, initial	1536 kW
Condensing fluid	Potassium
Potassium Flow Rate	1.80 lb/sec (.816 Kg/sec)
Condenser inlet quality	89%
Condenser inlet temperature, Pressure	1220°F, 5.4 psia (933°K, 37230 N/m^2)
Condenser outlet temperature, Pressure	980°F, 3.4 psia (800°K, 23440 N/m^2)
End-of-life heat rejection capability	75 percent initial (1152 KW)
Survival probability	0.99 & 0.999
Mission life	20,000 hrs. & 43,800 hrs. (7.2×10^8 sec & 15.4×10^8 sec)
Condenser material	Cb-1Zr
Radiator material	Stainless steel tube and armor; stainless steel clad copper or stainless steel fins; iron titanate coating, ε_t 0.90
Meteoroid criteria:	
Meteoroid density	0.5 gm/cm^3
Relative meteoroid velocity	25 km/sec
Constants for meteoroid equation	$N = \alpha m^{-\beta}$
α	$10^{-14.41}$ $gm^{1.22}/m^2 sec$
β	1.22
Launch vehicle	Two-stage Saturn V
Radiator sink assumptions	300 nm equatorial orbit; radiator's axis parallel to earth's surface; sun at zenith
Radiator supported load (see Figure 2-2)	15,000 lbs., 10° half-cone angle (6810 Kg)



NOTES: NO SHROUD ON RADIATORS.
15000# SYSTEM WEIGHT APPLIED AT "a-a" RING

FIGURE 2-2. SCHEMATIC OF RANKINE SYSTEM ON 2-STAGE SATURN V

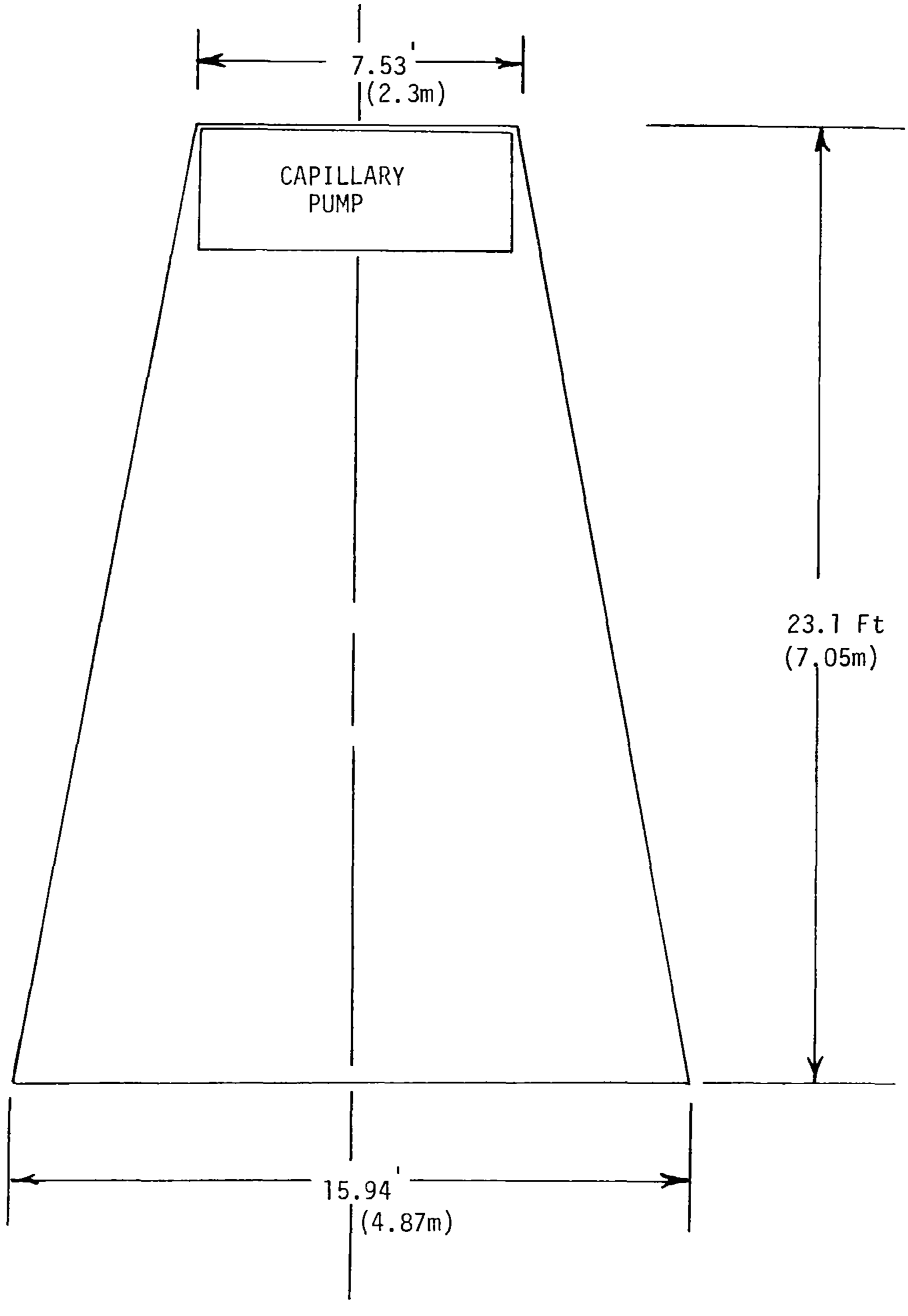


FIGURE 2-3. LOCATION OF THE CAPILLARY PUMPED HEAT REJECTION SYSTEM ON 2-STAGE SATURN V

3.0 CAPILLARY PUMP CONCEPTUAL DESIGN

The formulation of a conceptual capillary pump unit is a problem of designing a wick and arranging it relative to the heat source, liquid feed and vapor exit so as to obtain high thermodynamic and hydrodynamic efficiencies. This makes it possible to build compact pump units which are lightweight and will fit easily within the geometry available. High thermodynamic and hydrodynamic efficiencies are manifest by small temperature drop from the heat source and the wick/vapor interface, and low frictional pressure loss on the liquid side.

An early task in this study was the generation of a conceptual design and configuration for the capillary pump. The design concept selected was then to be used as the basis of a detailed computer program of the heat rejection system.

The starting point for this task was a careful analysis of the Stenger type pump. The detailed results of this analysis are presented in subsection 3.1, The Stenger Capillary Pump Design. A flat wick configuration, arterially fed liquid on the vapor side evolved as a product of the analysis of Stenger's design; this concept is analyzed in subsection 3.2, Capillary Pump with Conventional Wicks. Further consideration produced a concept employing a single screen wick with an artery liquid feed. The analysis of this concept is given in subsection 3.3, Capillary Pump with Screen Wicks.

3.1 The Stenger Capillary Pump Design

The Stenger-type design is shown schematically in Figure 3-1. The distinguishing feature of this design is that the liquid is supplied to the wick from one side, heat is supplied through the wall on the opposite side, and the vapor spaces are cut into the wick or wall at the interface between the hot wall and the wick. The working fluid is vaporized from the surfaces of these vapor spaces. Figure 3-1 shows the vapor spaces cut in the wall. Thus, the heat is transported to the wick through fins. A rectangular configuration of the same basic design but vapor spaces cut in the wick material is shown in Figure 3-2.

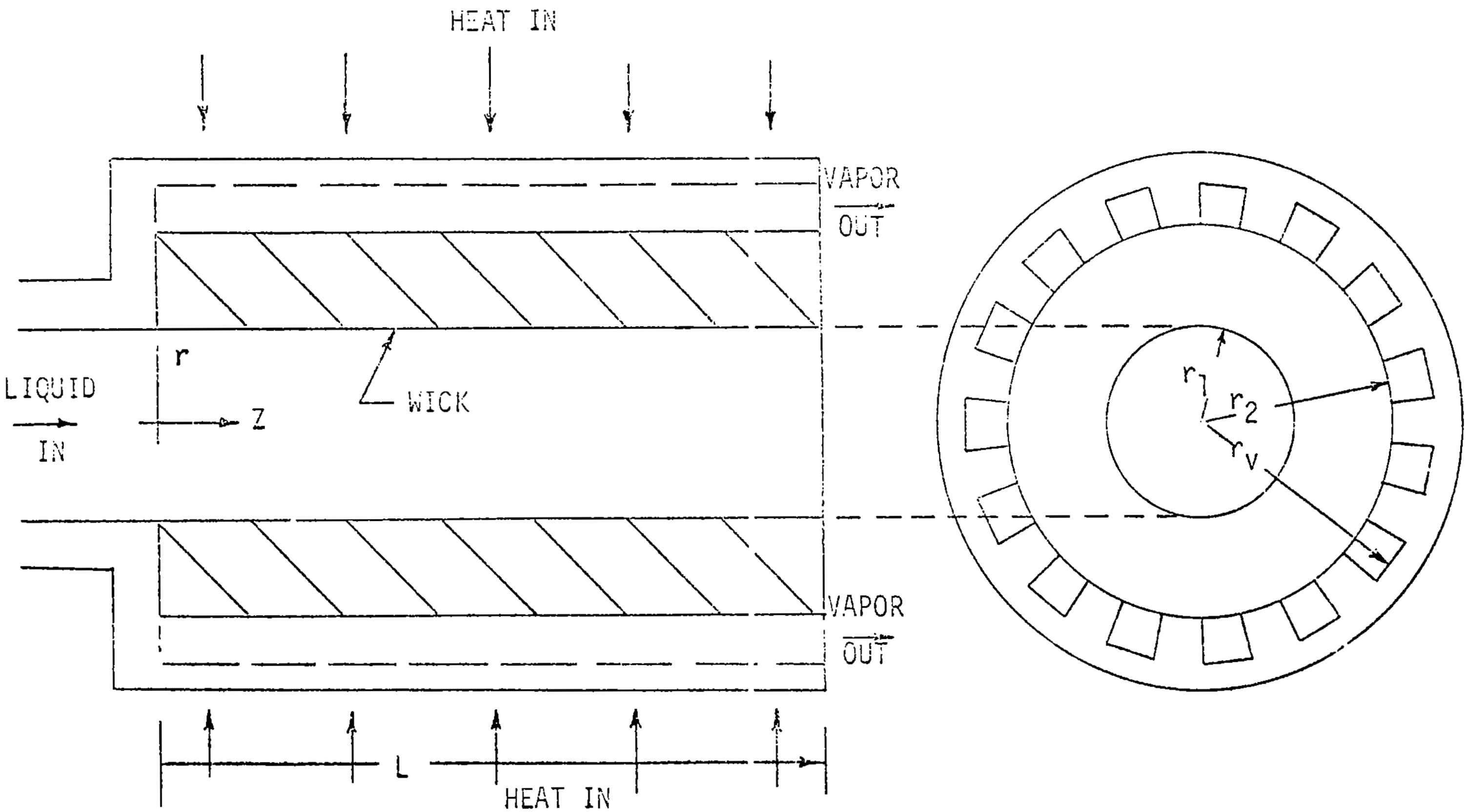


FIGURE 3-1. SCHEMATIC OF CYLINDRICAL CAPILLARY PUMP

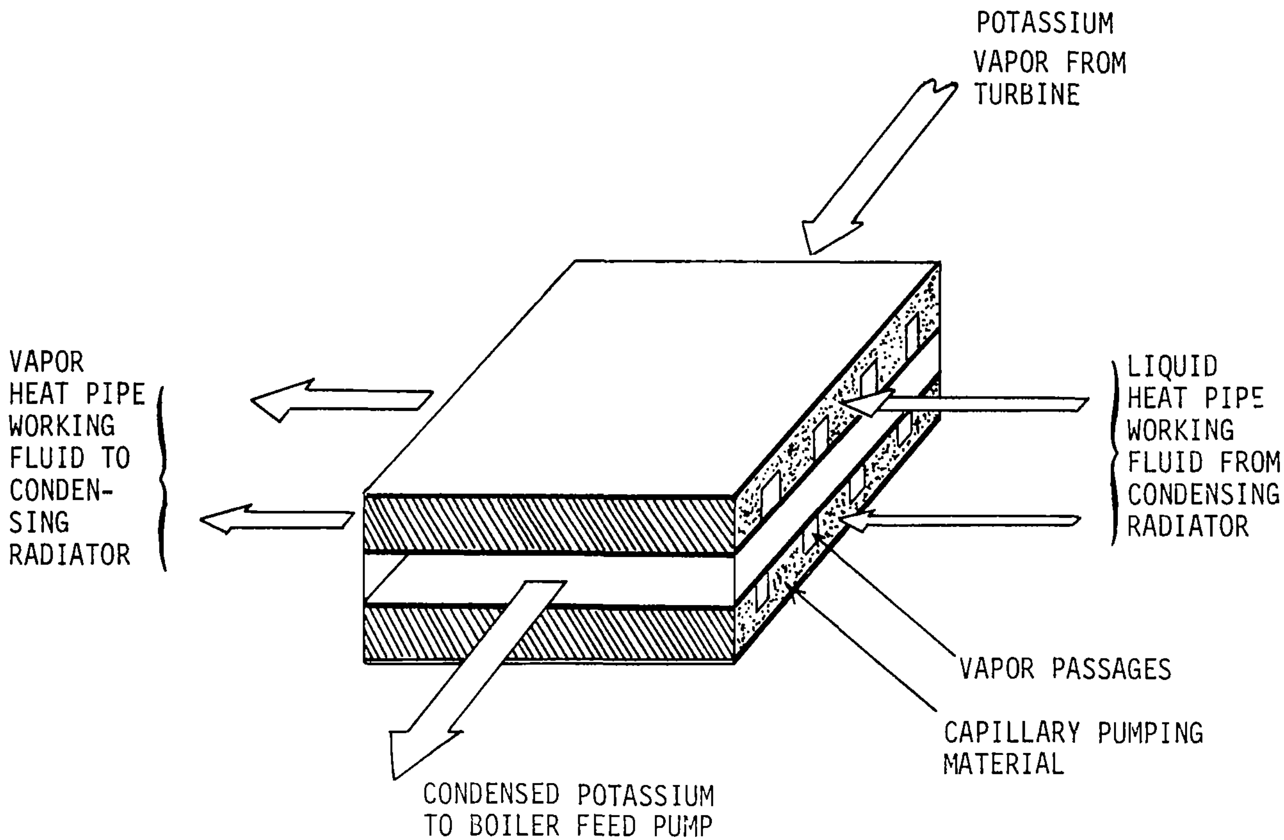


FIGURE 3-2. SCHEMATIC OF CAPILLARY BOILER

The cylindrical configuration shown in Figure 3-1 was subjected to an extensive analysis to determine the appropriate size of pump required to reject the total waste heat load imposed on the system and to determine the sensitivity of the size to changes in the system parameters. The details of this analysis are presented in section 1 of Appendix A, Analysis of Stenger Type Cylindrical Capillary Pump. This analysis optimized the pump by determining the pump dimensions which resulted in a minimum heat transfer area. This area was found to be inversely proportional to the square root of the fin thermal conductivity and the vapor flow velocity. Therefore, increasing the conductivity or the velocity results in a reduction of required heat transfer area. The analysis results showed that the vapor velocity at the exit of the vapor space was a major factor in limiting the axial length of the pump. Increasing the fin length or increasing the gap/fin ratio provides more vapor volume but at the expense of temperature drop which, in turn, adds substantially to the overall radiator weight. Therefore, to reject the total specified heat load would require a very large number of short pumpers; 755 for the sample case given in the Appendix. Further, to maintain a temperature drop across the wall of about 30°F(17°K), the total heat transfer area required is of the order 300 ft^2 ($28m^2$). This is five times the heat transfer area used in the single phase NaK heat exchangers in the present design (see Section 2.0, System Requirements). Since this limit is hydrodynamic, there is no advantage in cutting the vapor passages into the wick to create more wick/vapor area for vapor generation.

These calculations clearly show that vapor passages between the wick and the hot wall is inherently limited in capacity. This result, plus the packaging advantages in using a flat configuration instead of cylindrical, led to the pumper concept analyzed in the following subsection.

3.2 Capillary Pump with Conventional Wick

A "flat" configuration was derived which uses arteries to distribute the liquid on the vapor side of the wick. This arrangement is shown conceptually in Figure 3-3.

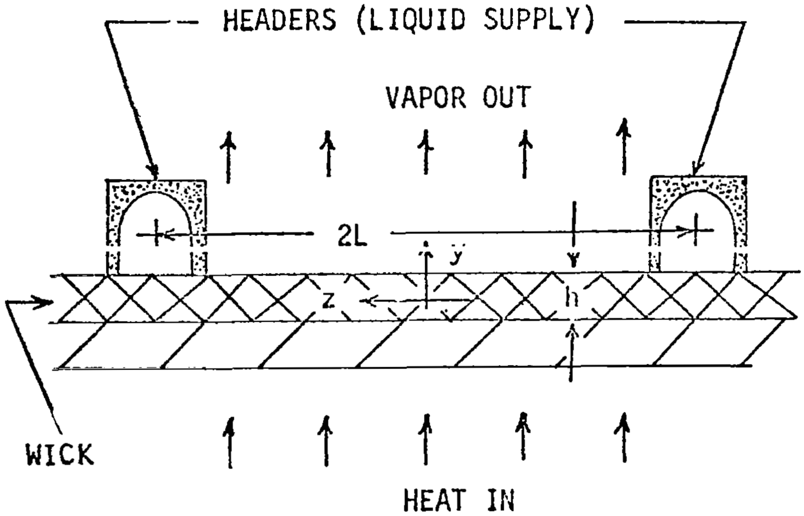


FIGURE 3-3. SCHEMATIC OF CAPILLARY PUMP CELL

This configuration was also analyzed in detail, the details being presented in Section 2 of Appendix A. The analysis, again, optimized the pump by determining the system dimensions which will reject the required heat load with a minimum heat transfer area. The wall to vapor temperature difference was limited to 30°F (17°K) to prevent nucleate boiling problems. The results showed that very thin wicks were required (~0.02 ft (0.006m) for typical materials) and that the total heat transfer surface area would be large compared to that required in the conventional EM-pumped heat rejection. For example, if the liquid feed arteries are separated by one foot, 185 ft^2 ($17.2m^2$) of wick surface area are required. Note, that the 30°F (17°K) temperature drop does not include that across the condensing potassium film or across the separating wall.

The results clearly show that a flat wick system with feed arteries on the vapor side gives much greater efficiency than the Stenger pump (nearly a 50% reduction in heat transfer area for comparable frictional pressure loss on the liquid side and temperature drop). The heat transfer area, however, is still three times larger than that required by the conventional system.

3.3 The Screen Wick Design

The capillary pumps described above use conventional fine pored wicks. Liquid flow through these wicks is a very dissipative process, and the pump is hydrodynamically inefficient. In order to remove the imposed heat load with the small temperature drop desired and low frictional pressure drop, fairly large heat transfer areas are required with liquid feed arteries closely spaced. To alleviate this problem, the composite-wick concept, widely used in heat pipe design, was applied to the capillary pump. This concept recognizes that it is only the fine-pored surface of the saturated wick which provides capillary pumping. Thus, only a two-dimensional wick is necessary. The third dimension, necessary for liquid flow, can be devoid of solid obstructions, thereby minimizing flow resistance.

The screen wick concept was analyzed in considerable detail to determine the required system dimension for achieving the system requirements. The details of these analyses are presented in section 3 of Appendix A.

In this analysis, the parameters of gap spacing, temperature drop and artery spacing were allowed to vary and the total heat transfer area required to remove the total heat load was calculated. The heat source temperature, the screen mesh (400), the working fluid (potassium) and the allowable pressure loss in the liquid were fixed. Figures A-9, A-10, and A-11 show the initial analysis results. These results appear very encouraging since they show that only about 30 ft^2 are required as compared to 185 ft^2 and 300 ft^2 for the two previous concepts under the same operating conditions. However, gap widths (wall to screen) of less than 10 mils were calculated. From a practical standpoint, it was felt that 15 mils spacing represented a lower value. Thus, the area required for the screen wick, for comparable temperature and pressure drops as used above, increases to about 80 ft^2 ($7.45 m^2$) (the present EM-pumped system utilizes 60 ft^2).

This result, however, shows that the screen wick is substantially more efficient than the other concepts considered.

An analysis was carried out to determine the performance improvement obtainable by tapering the screen-heated wall spacing away from the feed artery. An improvement in performance is expected because of the decreasing fluid flow rate. It was found that an optimum angle of only about 1/6 degree existed and that the improvement in performance was about 4%. The benefits are not compatible with the increase in fabrication problems and, consequently, further analysis was confined to the parallel screen geometry because of the inherent simplicity of manufacture.

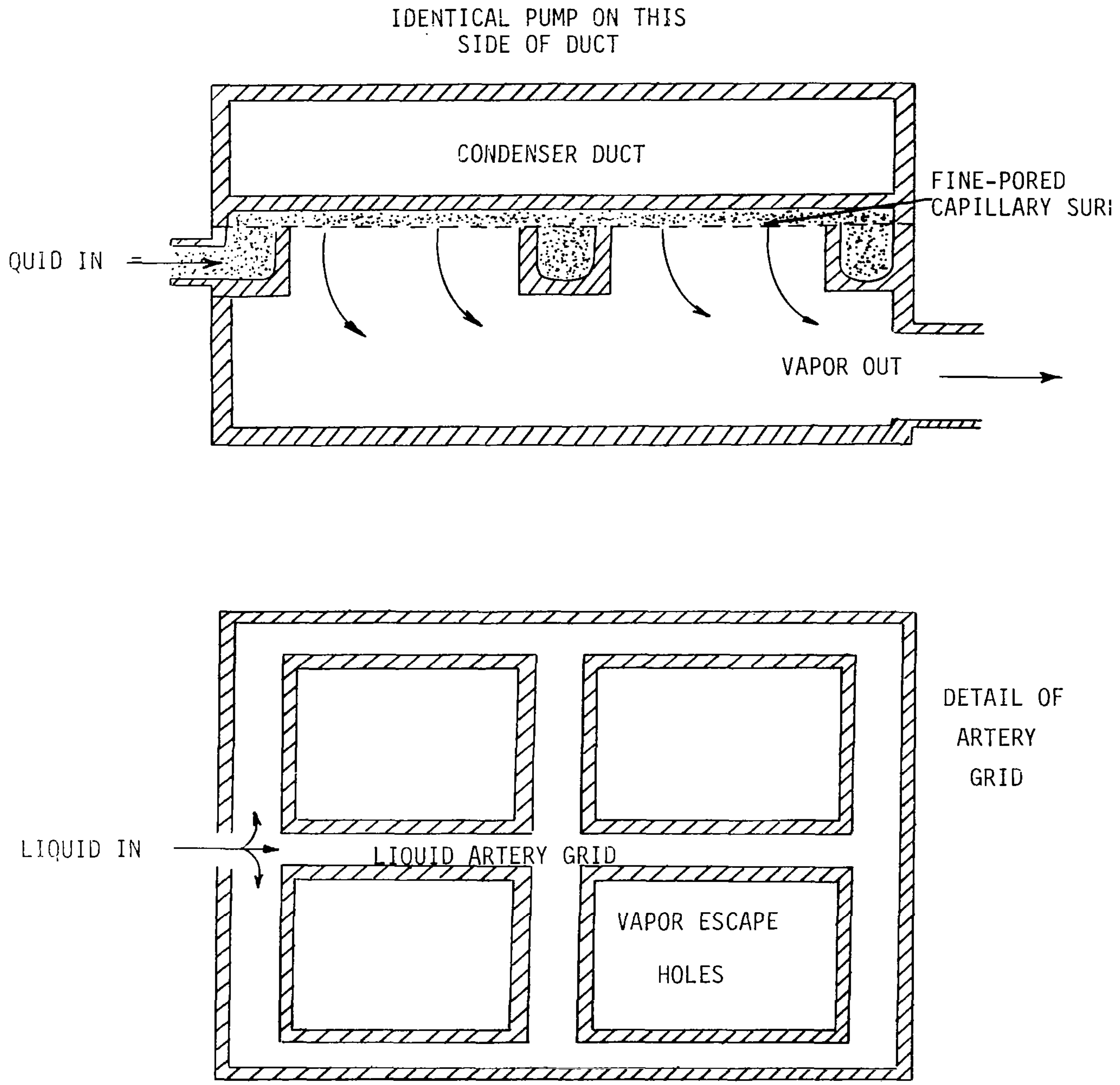


FIGURE 3-4. SCHEMATIC OF CAPILLARY PUMP UTILIZING BOTH ARTERIES AND COMPOSITE WICK

A word is in order at this point regarding the fabricability of the screen wick design concept. Referring to Figure 3-4, it is envisioned that the first unit could be made by machining out the liquid arteries and vapor escape holes in a solid metal sheet to form the artery grid. Fine screenwire ($\sim$ 400 mesh) would then be stretched over the face of the grid and the assembly sintered. Note that there must be no passages between the liquid feed arteries and the vapor space greater than the opening in the screen. Spacer nubs would be machined at appropriate intervals on the surface of the condenser duct to accurately maintain screen-heater surface spacing. The artery grid/screen assembly would then be positioned and welded to the surface of the condenser duct. Finally, the vapor chamber is welded to the artery grid. Structural support will be necessary to maintain dimensional constraints for the vapor chamber and the condenser duct.

Based on the results of the analyses described above and a cursory examination of the problems of fabrication, it was concluded that the screen wick design offers substantial advantages over the other concepts studied with respect to performance, temperature difference, required area and probably overall weight. In addition, the inherent simplicity offers obvious fabrication advantages. Consequently, the screen wick design was selected for the detailed system design and performance studies described in the following sections.

4.0 SYSTEM ANALYSIS MODEL

This section of the report presents a description of the system analysis method, including the equations used to analyze each system component, the assumptions made in deriving and solving these equations and the boundary conditions required for their solution.

4.1 CAPILLARY PUMP MODEL

A physical model of the capillary pump and the heat rejection system was derived as a result of the conceptual design studies reported in Section 3. This model, which forms the basis of the computer programs developed, is described in this section.

The basic configuration of the capillary pump is that shown in Figure 3-4 and Figure 4-1. These drawings show a long, narrow rectangular potassium condenser channel with pumps attached to the long sides. The pump is a rectangular chamber into which a network of arteries is built to carry the working fluid to the screen wick. The pump possesses the following key features.

A screen is placed a fixed distance from the condenser channel wall. Liquid is fed into the artery grid from one side (the bottom of Figure 4-1) where it flows into the space between the hot wall and the screen. The vapor generated at the screen flows to the exit ports where it enters a collection manifold and is transported to the condensing radiator.

The dimensions of the pumper which enter the design program as free parameters are the width between the hot surface and the screen (gap width), the half distance between arteries (element half width), the heat transfer area and the radius of the holes in the screen (capillary effective radius). The design program was used to determine the value of these parameters which resulted in a minimum heat rejection loop weight.

The screen wick must be supported during the pump operation because of the small pressure difference across it. This support is envisioned to be given by a grid work of nubs on the hot potassium wall or by a very course

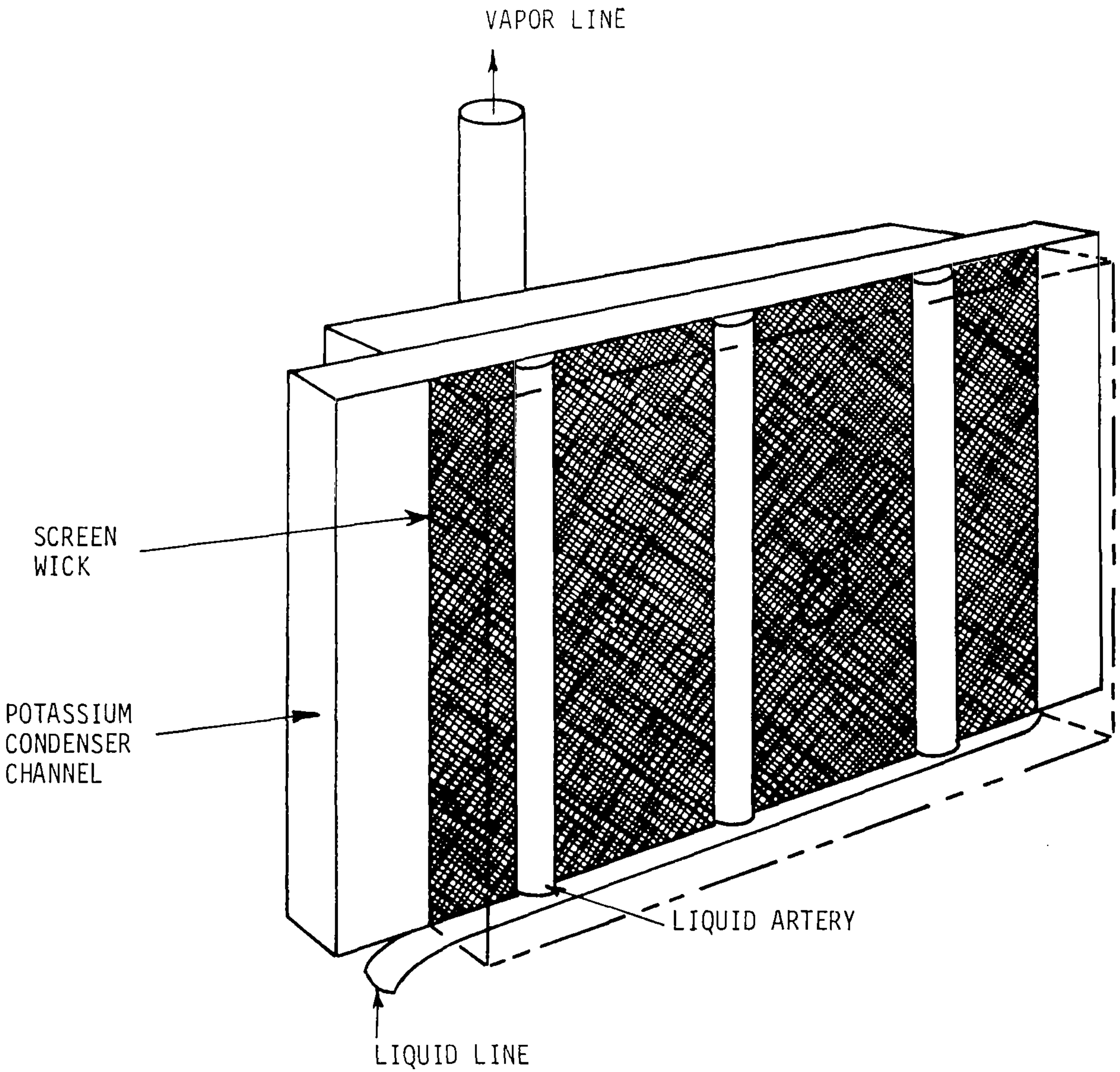


FIGURE 4-1. CAPILLARY PUMP PHYSICAL MODEL

screen of controlled thickness, placed behind the wick. The first method gives lower pressure drop for flow behind the screen, but is probably more difficult to manufacture. Both methods appear feasible.

Initially, it was supposed that each pumper straddle the potassium condenser channel and the vapor generated on each side was mixed in a common header before being transported to the radiator. The performance program was written on this supposition so that the heat rejection loops were in series. Thus, the state properties of the condensing potassium leaving loop n were the input conditions for loop $n+1$. Later, it was found that some simplification of the system plumbing could be realized by having each loop take heat from only one side of the condenser channel In this case, each two loops are in parallel. For example, loop 1 and 2 are subjected to the same condensing potassium conditions. A method had to be derived whereby the series-parallel combination of loops performance could be obtained from a pure series calculation. This method will be discussed in Section 5.

4.2 CAPILLARY PUMP EQUATIONS

Consider Figure 4-2, which shows a schematic of the capillary pumped heat rejection loop. The circled numbers are points where the state of the working fluid is to be determined. Point 1 is the vapor exit of the pump. Region 1 → 2 is the vapor line leading to the radiator. Point 3 is the exit of the radiator and region 3 → 4 is the liquid return line from the radiator to the pumper.

The equations must be solved for two types of problems, (1) a design solution and (2) a performance solution. The design program begins with a specific pump design, imposed source and sink temperatures and heat input rate and calculates the corresponding optimum (minimum weight) radiator dimensions. The performance program begins with a given loop design (both pump and radiator), imposed source and sink temperatures and calculates performance of a series of identical capillary pumped loops attached to the source.

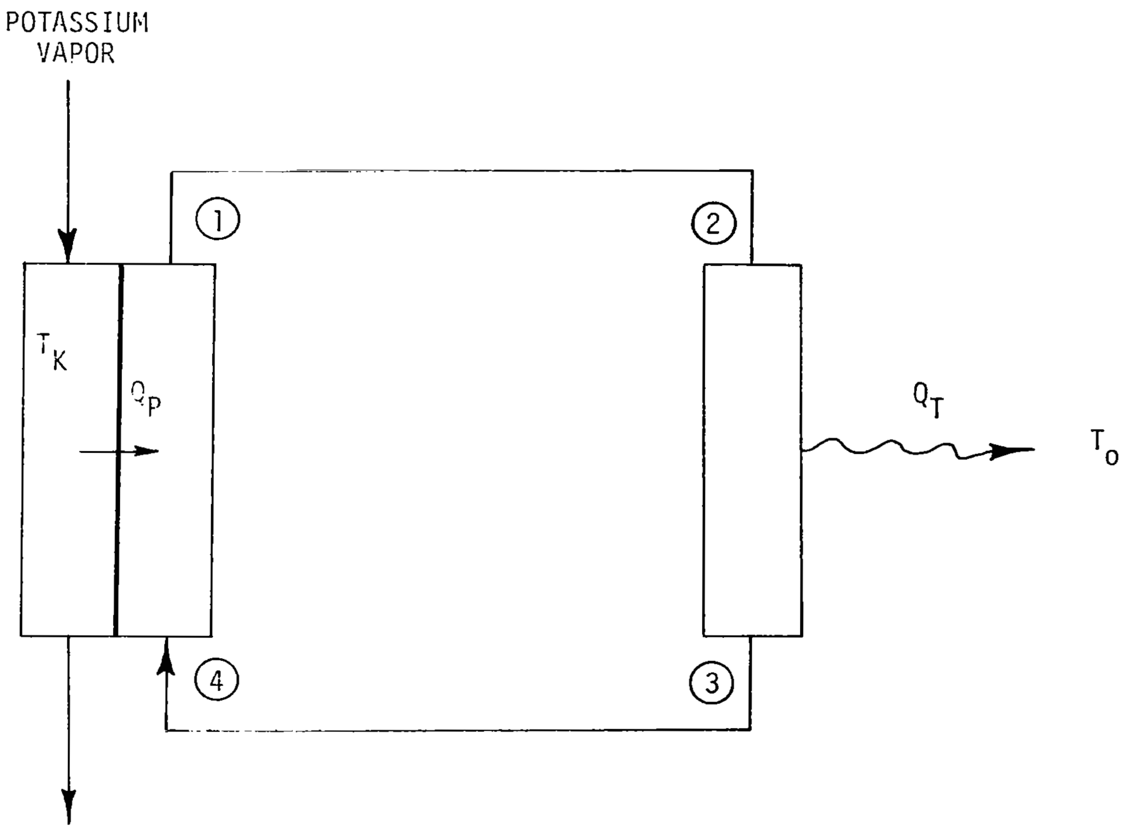


FIGURE 4-2. CAPILLARY PUMPED HEAT REJECTION LOOP SCHEMATIC

The thermodynamic and physical properties are functions of temperature in the analysis. The equations which define these functionalities are presented in Appendix B.

Since the analyses to be performed are steady state, the presentation to follow can be simplified by giving the overall conservation of mass equation as

$$W_i = W = \text{Constant} \qquad 4\text{-}1$$

This states that the mass flow rate at any point (i) in the system is constant. The equations to follow are written with one flow rate, W.

Heat, given up by the condensing potassium, is conducted through the condenser wall, working fluid film and screen wick to the liquid-vapor interface. The interface temperature is spatially constant and working fluid is neglected. Thus, the process is just condensation and conduction and can be represented by

$$Q_p = A_p \, U(T_K - T_1) \qquad 4\text{-}2$$

The heat transfer coefficient

$$\frac{1}{U} = \frac{1}{h_c} + \frac{\Delta X_w}{k_w} + \frac{\Delta X_f}{k_f} \qquad 4\text{-}3$$

The energy conservation equation is, simply,

$$Q_p = W(h_1 - h_4) \qquad 4\text{-}4$$

The variable h_1 is the saturation enthalpy of the working fluid vapor at T_1 and thus these two variables are related by a thermodynamic equation from Appendix A.

The highest pressure in the system is P_1 which is the saturation pressure of the working fluid at the temperature T_1. These variables are also related by the vapor equations from Appendix B.

The momentum equation for flow through the pumper is

$$P_1 - P_4 = \frac{2\varsigma}{R} K - \left[\frac{1}{\rho_1 A_1^2} - \frac{1}{\rho_4 A_4^2} \right] \frac{W^2}{g_c} - \frac{6\mu k\ \Delta T^2\ L^2}{\rho (X_f + X_s) X_f^3 \lambda\ g_c} \qquad 4\text{-}5$$

The first term is the pressure rise due to capillary forces which will adjust, to a limit of K=1, to match the pressure losses in the system. The second term is the pressure loss due to momentum gain of the working fluid upon evaporation and the last is the friction pressure drop in the liquid flow.

4.3 VAPOR TRANSPORT LINE

The vapor line is assumed to be adiabatic and therefore the conservation of energy equation states that

$$T_1 = T_2 \qquad 4\text{-}6$$

The momentum equation gives the pressure drop in this line as,

$$\Delta P_{1 \to 2} = f \left(\frac{2L_v}{D} \right) \frac{1}{\rho_v} \left(\frac{W}{A_v} \right)^2 \frac{1}{g_c} \qquad 4\text{-}7$$

where the friction factor assuming turbulent flow is given by Blasius equation,

$$f = \frac{0.0791}{\left(\frac{WD}{A\mu} \right)^{0.25}} \qquad 4\text{-}8$$

Substitution of equation 4-8 into 4-7 gives

$$\Delta P_{1 \to 2} = 0.263(10^{-11}) \left(\frac{\mu_v}{\rho_v} \right)^{0.75} \left(\frac{W}{A_1} \right)^{1.75} \left(\frac{L_{12}}{D_1^{1.25}} \right) \qquad 4\text{-}9$$

4.4 LIQUID TRANSPORT LINE

The liquid line is also assumed to be adiabatic and,

$$T_3 = T_4 \tag{4-10}$$

The pressure drop through the liquid line is given by an equation similar to equation 4-9, again assuming turbulent flow.

It is,

$$\Delta P_{3 \rightarrow 4} = 0.0379 \ (10^{-8}) \ \frac{\mu_L^{0.25}}{\rho_L} \left(\frac{W}{A_4}\right)^{1.75} \frac{L_{34}}{D_4^{1.25}} \tag{4-11}$$

4.5 RADIATOR/CONDENSER

4.5.1 Heat Transfer Analysis

Heat rejection in the radiator involves convection from the fluid to the tube wall, conduction from the tube to the fins and radiation from the tubes and fins to space.

Heat Transfer Equations

The analysis technique in the radiator involves dividing the radiator into ten longitudinal increments. Heat balance equations are solved to determine the outlet conditions of each incremental element for known inlet conditions. Calculation starts at the condenser inlet and marches to the outlet. A representative element is shown in Figure 4- 3.

The radiation fin effectiveness is utilized to account for temperature drop from the tube to fin. Heat transfer in the direction parallel to the tubes is neglected, and the inner surface of the radiator is assumed to be insulated.

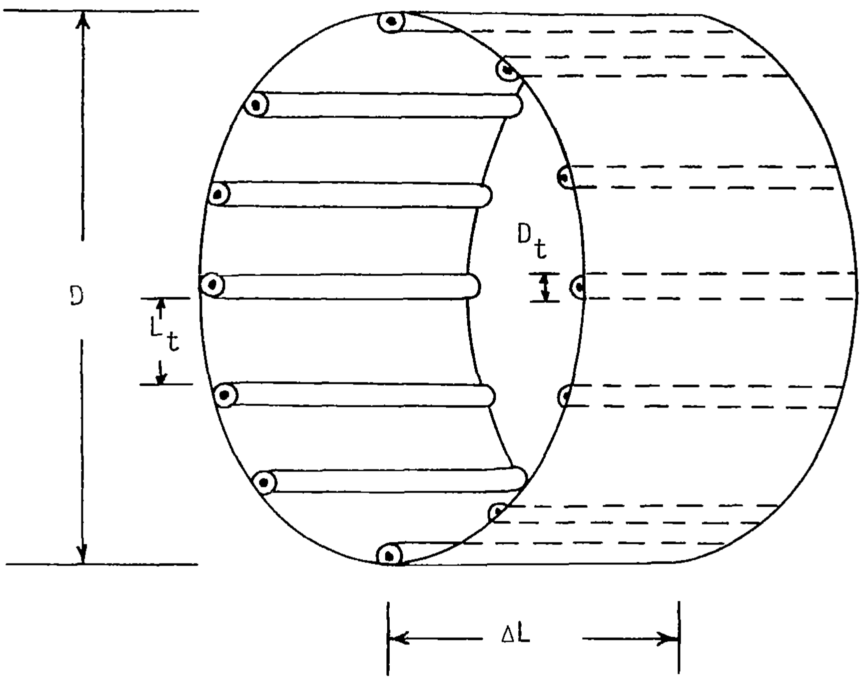


FIGURE 4-3. RADIATOR ELEMENT

Heat transfer from a radiator element can then be expressed as:

$$Q = \varepsilon\cdot\sigma\cdot D_t\cdot N_t\cdot\Delta L\left(T_w^{\,4} - T_{sk}^{\,4}\right) + \varepsilon\cdot\sigma\cdot L_t\cdot N_t\cdot\Delta L\cdot\eta\left(T_w^{\,4} - T_{sk}^{\,4}\right) \qquad 4\text{-}12$$

where T_w and T_s are the tube wall and sink temperatures, respectively, and η is the mean fin effectiveness for the element. Fin effectiveness tables are included as part of both the design and performance computer programs.

The first term is heat transferred from the fin area under the tube which is assumed to have an effectiveness of one. The fin effectiveness is determined for each element and assumed to be uniform over the element. Temperature of the wall is assumed constant for each element.

The heat transfer by radiation is equal to that transferred to the tube inner wall by convection:

$$Q = h\cdot A_t\cdot\ (T_f - T_w) \qquad 4\text{-}13$$

In the condensing section, the fluid temperature over an element is assumed constant and sensible heat transfer is neglected. The heat transfer from the fluid is equal to:

$$Q = W\cdot\lambda\ (X_{in} - X_{out}) \qquad 4\text{-}15$$

In the subcooler section, the heat transfer is expressed by:

$$Q = WC_p\ (T_{in} - T_{out}) \qquad 4\text{-}15$$

Heat Transfer Coefficients

Condensing heat transfer coefficients were input as constant for each fluid. Values used were:

Potassium = h_c = 10,500 Btu/hr-ft^2°F (Reference 5)

Sodium & Cesium = h_c = 3400 Btu/hr-ft^2°F (Reference 6)

Resistance of the liquid film was neglected in condenser section heat transfer calculations. Assuming a liquid-vapor velocity ratio of one, the following Table 4-1 gives the conductance of the liquid layer as a function of local fluid quality, X, for potassium.

Table 4-1. Film Conductance Versus Quality

X	Film Conductance (Tube D=1"), Btu/hr ft^2°F	(kw/m^2 °K)
1.0		
.75	60×10^5	34×10^3
.5	20×10^5	11.3×10^3
.1	2.2×10^5	1.25×10^3
.05	1×10^5	$.57 \times 10^3$
.01	0.1×10^5	$.057 \times 10^3$

As can be seen from Table 4-1, the film conductance becomes of the same order of magnitude as the condensing heat transfer coefficient (10^4) only very near the end of the condensing section; ie., small quality, and therefore neglecting the film resistance introduces a small error.

The heat transfer coefficient in the radiator sub-cooler section is based on a correlation of Nusselt number versus Peclet number as recommended by Reference 7.

$$Nu = 0.625\ Pe^{.4} \qquad 4\text{-}16$$

Sink Temperature

Direct solar, earth albedo and earth thermal heat inputs to the radiator exterior surface, averaged over one orbit, are approximately 120 Btu/hr-ft^2 (0.378 $\frac{kw}{m^2}$) a 300 n.m. earth equitorial orbit was assumed.

This gives an equivalent sink temperature for the radiator of 68°F (293°K) which was used as input to the optimization and performance program. At the nominal radiator temperature (1200°F, 922°K) the influence of this sink is negligible.

4.5.2 Radiator Pressure Drop Calculations

The overall static pressure change between inlet and outlet of the radiator is subdivided as follows:

1. Inlet header frictional pressure loss
2. Header-to-radiator turning and entrance loss
3. Two phase frictional condensing pressure drop
4. Pressure rise due to momentum recovery of condensed liquid
5. Frictional loss in liquid subcooling section.
6. Tube to exit header turning loss
7. Exit header frictional pressure loss

Inlet Header Frictional Loss

The headers are designed internal to the computer programs such that the vapor velocity is constant throughout the header. The velocity used in the header design is the same as that at inlet of the radiator tubes. Pressure drop is calculated from the following equation using the average header diameter.

$$\Delta P = \frac{0.316}{Re^{.25}} \cdot \frac{\rho V^2}{2g} \cdot \frac{L}{D} \qquad 4\text{-}17$$

Turbulent flow is assumed with no check of the Reynolds number.

Inlet Turning and Entrance Losses

Entrance and turning losses are assumed to be equal to one velocity head.

Two Phase Pressure Drop

In analyzing two-phase frictional pressure drop, it is convenient to introduce the Lockhart-Martinelli frictional pressure drop modulus, defined as:

$$\phi^2 = \frac{dP/dL)\ \text{Two phase}}{dP/dL)\ \text{Vapor only}} \qquad 4\text{-}18$$

The quantity, ϕ^2, is a measure of the influence of the liquid phase on the loss in pressure due to friction. Accepting this correlation, the problem reduces to determining the value of ϕ^2 consistent with the existing two-phase flow pattern.

A rough wall model was developed for the turbulent-turbulent regime (8). Comparisons of this model with test results for potassium are given in Reference 13. The correlation involves an experimental correction of the original Lockhart-Martinelli correlation using Refrigerant 11. Formulation of the correlation is:

$$\phi^2 = \frac{Ce}{.046} \cdot \phi_g^2 \qquad 4\text{-}19$$

Where the relation for Ce is

$$\begin{aligned} Ce &= 0.1, && \theta \leq 57.23 \\ Ce &= 4.182/\theta^{.9225}, && 57.23 < \theta < 327.59 \qquad 4\text{-}20 \\ Ce &= 0.02, && \theta \geq 327.59 \end{aligned}$$

Where $\theta = \dfrac{\zeta \cdot g_c}{\mu_v V}$

ϕ_g is a function of the quantity, K, below.

$$K = \left[\left(\frac{w_f}{w_v}\right)^{1.8} \left(\frac{\rho_v}{\rho_e}\right)\left(\frac{\mu_e}{\mu_v}\right)^{.2}\right]^{1/2} \qquad 4\text{-}21$$

The ϕg v.s. K relation is given in Figure 4-4 (8). Investigation of flow regimes in typical optimum weight radiators shows that in all cases the vapor flow is turbulent. The film Reynolds number is in the laminar regime for a substantial length of the radiator. The above correlation has been utilized although it is recognized that in most of the radiator the film Reynold's number is below that where experimental verification is available.

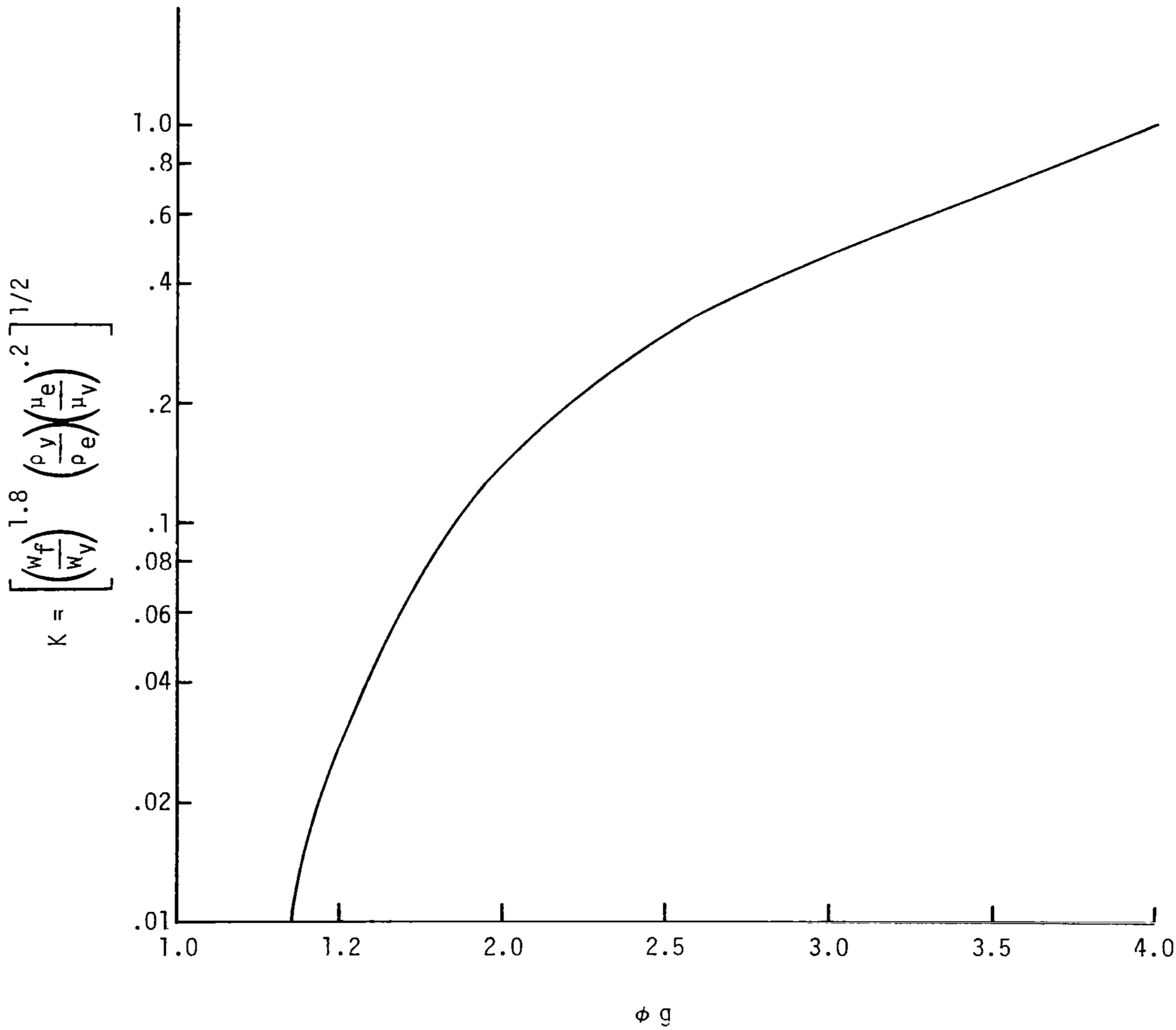


FIGURE 4-4. LOCKHART-MARTINELLI PRESSURE DROP CORRELATION FOR TWO PHASE FLOW

In the computer programs developed, the condenser section is divided into ten increments. The pressure drop is calculated for each increment using local fluid conditions and the total two-phase drop is calculated by summing all individual increment pressure drops.

4.5.3 Pressure Rise Due to Momentum Recovery

There is essentially no liquid present at the condenser inlet ($X = 1.0$) and at the condensation interface the liquid velocity is assumed zero; therefore, there is no change in liquid momentum. Since there is no vapor momentum at the interface (zero velocity and flow rate) the momentum pressure recovery becomes:

$$\Delta P_{mom} = \left(\frac{\rho V_{in}^{2}}{g}\right)_{vapor} \tag{4-22}$$

4.5.4 Liquid Phase Pressure Drop

Frictional losses in the subcooler and exit header where liquid phase exists, have been neglected because of the low velocities and short flow path involved.

4.6 METEOROID PROTECTION

Calculation of required tube wall thickness to prevent meteoroid puncture requires relations for penetration thickness and specification of the meteoroid environment. The following penetration equation was adopted (9):

$$t = (\gamma)(a)(d)\left(\frac{\rho_m}{\rho_t}\right)^{1/2}\left(\frac{V}{c}\right)^{2/3}\left(\frac{1}{1+\beta}\right)^{1/3\beta} \tag{4-23}$$

where the terms are explained in Appendix G.

Specification of the meteoroid environment in addition to the velocity and density must include a relation between the flux density and mass,

$$N = \alpha \cdot m^{-\beta} \qquad 4\text{-}24$$

where N is the flux density of meteoroids with mass greater than m. The constants α and β are specified as

$$\alpha = 10^{-14.41} \ \text{gram}^{1.22}/\text{m}^2\text{-sec}$$
$$\beta = 1.22$$

The above relations must be combined with a probability equation. Assuming a Possion distribution:

$$P(N \leq N^1) = \sum_{x=0}^{x=N^1} \frac{e^{-n\cdot p}(n\cdot p)^x}{x!} \qquad 4\text{-}25$$

The quantity $n\cdot p$ can be determined in terms of the mass distribution equation as:

$$n\cdot p = \alpha\cdot M_{crit}^{-\beta}\cdot\tau\cdot A \qquad 4\text{-}26$$

M_{crit} = Minimum mass which will give penetration

τ = Mission time

A = Vulnerable area

Equation 4-26 can be used to eliminate the meteoroid diameter from equation 4-23 and allow calculation of the thickness required for a given number of penetrations and probability. The meteoroid diameter which must be designed for becomes:

$$d = \left(\left(\frac{\alpha\ \tau\ A}{np}\right)^{\frac{1}{3\beta}}\left(\frac{6}{\pi\rho_m}\right)\right)^{1/3} \qquad 4\text{-}27$$

The final penetration equation becomes:

$$t = (\gamma)(a)\left(\frac{\alpha\cdot\tau\cdot A}{n\cdot p}\right)^{1/3\beta}\left(\frac{6}{\pi\rho_p}\right)^{1/3}\left(\frac{\rho_m}{\rho_t}\right)^{1/2}\left(\frac{v}{c}\right)^{2/3}\left(\frac{1}{1+\beta}\right)^{1/3\beta} \qquad 4\text{-}28$$

Assuming a stainless steel armor material, the following relation for thickness required in terms of mission time, vulnerable area and the quantity $n\cdot p$ results:

$$t = 9.35\times10^{-3}\left(\frac{\tau A}{n\cdot p}\right)^{0.273} \qquad 4\text{-}29$$

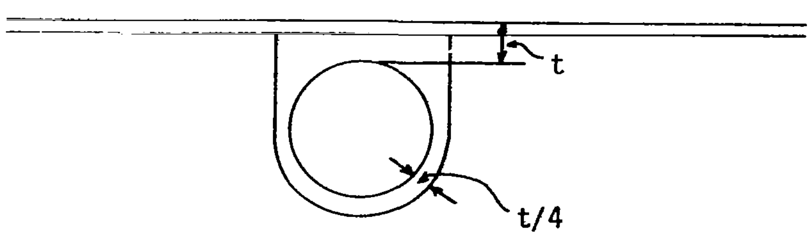
Where t is in cm, τ in days and A in square meters.

The quantity np is a function of the number of penetrations allowable and the survival probability. This data is obtained from a solution of equation 4-25 and is summarized in Table 4-2 below.

Table 4-2. n·p Relation

Number of Penetrations	P(N) = 0.99	P(N) = 0.999
0	.010	.001
1	.150	.048
2	.430	.195
3	.830	.430
4	1.28	.70
5	1.80	1.12

Shielding configuration assumed is shown below.



This configuration allows a reduction in side wall thickness because of the scattering effect of the fin surface. A reduction by a factor of four was adopted from Reference 9.

5.0 SYSTEM PRELIMINARY DESIGN

This section describes and gives the results of studies made in formulating a preliminary heat rejection system design. These studies involved the application of the design and performance programs to select a working fluid, number of heat rejection loops and individual loop size to reject the imposed 1536 Kw heat load. The system design was also analyzed to gain information on its performance when subjected to off design conditions.

Subsection 5.1, Design Studies, presents the results of those studies in formulating a specific capillary pumped heat rejection system design. Subsection 5.2, System Conceptual Design, gives a quantitative summary of the system design. Subsection 5.3, Performance Studies, gives the normal and off design performance of the system.

5.1 DESIGN STUDIES

Studies were made to determine the best specific design for a complete capillary pumped heat rejection system. These studies employed both the design and performance computer program and a comparative evaluation of design factors, such as fluid properties, manufacturability, reliability and controlability. The results of these studies are presented in this section.

5.1.1 Working Fluid Selection

The evaluation of working fluids for capillary pumped systems must consider several factors. As a minimum, these include its chemical reactivity properties, its heat transporting and capillary pumping capability, its vapor pressure at the operating temperature, and the heat rejection system weight. The potential working fluids, sodium, potassium, and cesium, are evaluated below in light of these criteria to establish which has the most desirable properties.

The first requirements of a working fluid for capillary pumped systems are that it wet the wick material and be non-corrosive to the wick and other materials of construction. It is also desirable, for safety sake, that the fluids are relatively inert. All three potential working fluids wet stainless steel well. The corrosive properties are likewise comparable. Corrosion data indicate that most metals resist corrosive attack by the molten alkali metals, providing that they are oxide-free (10, 11).

The alkali metals are all very reactive materials. In order of decreasing reactivity they are cesium, potassium, and sodium. Table 5-1 gives a comparison between the reactivities of the three metals with some common inorganic materials (10). The metals are so reactive that they must all be handled with great care and the heat rejection loops will have to be completely free of oxygen. There is a slight perference of sodium, since it is slightly less reactive.

Analyses of capillary pumped systems show that the pumping capability of a working fluid is proportional to the property group (12).

$$N = \frac{\rho \lambda \zeta}{\mu} \qquad 5\text{-}1$$

Table 5-2 gives the properties of the three fluids at 1200°F (922°K). These data show that sodium is the far superior working fluid with regard to its pumping capability. The value of the property grouping for sodium is four times greater than that for potassium and sixteen times greater than that for cesium. The primary reason for this advantage is that sodium has a much larger heat of vaporization and surface tension than the other two fluids.

In summary, the three potential working fluids can be ranged according to their capillary pumping ability as sodium, potassium, and cesium in descending order.

Table 5-1. Chemical Reactions of the Alkali Metals

With	Sodium	Potassium	Cesium
Oxygen	Fairly rapid	Fairly rapid	Burn in air
Nitrogen	No reaction	No reaction	No reaction
Hydrogen	Rapid reaction above 300°C	Rapid reaction above 300°C	Reacts slowly at 600°C
Water	Rapid	More rapid	Most rapid
Carbon	Reacts at 800-900°C to give Na_2C_2	Dissolves to solid solution; no carbide formed	No carbide formed
NH_3	Reacts to give $NaNH_2$ (slow)	Reacts to give KNH_2 (easy)	Reacts to ive $CsNH_2$ (most rapid)
CO	No carbonyl formed, except in liquid NH_3	Forms explosive carbonyl	Absorbs CO at room temperature
CO_2	Reacts	Reacts	Most rapid reaction
Halogens:			
F	Ignites	Reacts violently	Reacts most vigorously of all alkali metals
Cl	Reacts	Reacts violently	
Br	Slow reaction	Detonates	
I	No reaction	Reacts, ignites	
H_2SO_4:			
Cold, conc.	Fairly vigorous	Explosive reaction	Explosive reaction
Cold, dilute	Very vigorous	Explosive reaction	Explosive reaction

Table 5-2. Properties of Alkali Metals

	$\lambda\ \frac{cal}{gm}$ $\left(\frac{joule}{Kgm}\right)$	$\rho\ \frac{gm}{cm^3}$ $\left(\frac{Kgm}{m^3}\right)$	$\zeta\ \frac{dynes}{cm}$ $\left(\frac{Newtons}{meter}\right)$	$\mu\ \frac{dyne\ sec}{cm^2}$ $\left(\frac{Newton\ sec}{m^2}\right)$	$N\ \frac{Kw}{sec}$
Sodium	984.58 (4119483)	0.798 (798.)	138.15 (0.13815)	0.001975 (0.0001975)	230,000
Potassium	460.27 (1925770)	0.691 (691.)	73.35 (0.7335)	0.001475 (0.0001475)	66,000
Cesium	117.67 (492331)	1.495 (1495.)	42.89 (0.04289)	0.001716 (0.0001716)	18,400

It is desirable for the temperature difference between the heat source and the pump vapor temperatures to be small for two reasons. First, the higher the vapor temperature, the more efficient the radiator and the more pressure available to generate flow and second, the amount of superheat at the hot surface that can be tolerated is limited to about 100°F (38°K). Thus, the best fluid is one which has a high thermal conductivity. Table 5-3 compares the thermal conductivities of the working fluids at 1200°F (922°K).

Table 5-3. Thermal Conductivity of Alkali Metals (1200°F, 922°K)

	k cal/cm sec °K	$\left(\frac{w}{m^2 °K}\right)$
Sodium	35.55	(14874.)
Potassium	19.55	(8180.)
Cesium	9.00	(3766.)

The data in this table shows that sodium is the preferable working fluid, followed by potassium and then cesium.

The vapor pressure of the working fluid at the pump temperature is the total driving pressure which generates flow. This vapor pressure must be sufficient so that it is greater than the sum of all the pressure drops around the loop. A practical heat rejection loop design will have about one-half psi total pressure loss. This can be reduced by increasing the tube diameters leading to and from the radiator and in the radiator. However, the size of the radiator becomes excessive. Table 5-4 gives a comparison of the vapor pressures for the three fluids at several temperatures within the range of interest to this study. The data show that at the highest temperature of 1200°F (922°K) all the fluids have a high enough vapor pressure to produce flow, although sodium is marginal at 1 psi (7000 N/m^2). On the other hand, at the lower temperature, between 900 (755) to 1000°F (811°K), sodium is not an acceptable fluid because its vapor pressure is too low. Under the ground rule that each pump should be designed to be identical, sodium can be eliminated as a potential working fluid in this application.

Table 5-4. Vapor Pressure of Alkali Metals in PSIA (N/m^2)

Temperature °F (°K)	1200 (922)	1100 (866)	1000 (811)	900 (755)
Sodium	1.04 (7170)	0.449 (3096)	0.173 (1193)	0.058 (400)
Potassium	4.66 (32129)	2.36 (16272)	1.19 (8205)	0.451 (3110)
Cesium	12.03 (82943)	6.67 (45988)	3.42 (23580)	1.59 (10963)

The design program was used to determine which of the three fluids resulted in the lightest system. These calculations will be expanded upon in the following section, but the result is presented here so that all factors are available for selecting a fluid. It was found that, under all conditions, potassium resulted in the lowest weight system. The reason for this result is that sodium, due to its low vapor pressure, and cesium, due to its low capillary pressure rise, require large flow passages in the radiator to minimize the pressure drop.

An added bonus with potassium is that the pressure difference across the condenser wall will be small since potassium is used in the power loop. It will, therefore, be easier to prevent distortion of the condenser wall and to maintain geometrical tolerances in the wide-wall separation.

The conclusions are that (1) sodium is not an acceptable working fluid for this application because its vapor pressure at the lowest temperature, about 950°F,(783°K), is too low to support the flow: (2) therefore, potassium is the chosen working fluid. It ranked second in its pumping capacity and in its thermal conductance, and possessed sufficient vapor pressure to maintain flow even at the lowest temperature; (3) cesium's only strong point is its high vapor pressure but its poor transport properties make it inferior to potassium. The high vapor pressure relative to potassium would also result in thicker and heavier condenser wall.

5.1.2 The Effect of Design Parameters Upon System Weight

The design program was used to determine the minimum weight capillary pumped system by varying each of the pump dimensional design parameters in turn, calculating the corresponding radiator weight and dimensions. The weight data were plotted and the minimum weight system determined out of core.

The pump design parameters input to the program are: the wick area, the width of the gap between the screen wick and the heat transfer wall, the half distance between the feed arteries. The heat rejection system parameters are the number of loops and the working fluid. The number of capillary loops varied in multiples of four, that is, 4, 8, 12, 16, and 20.

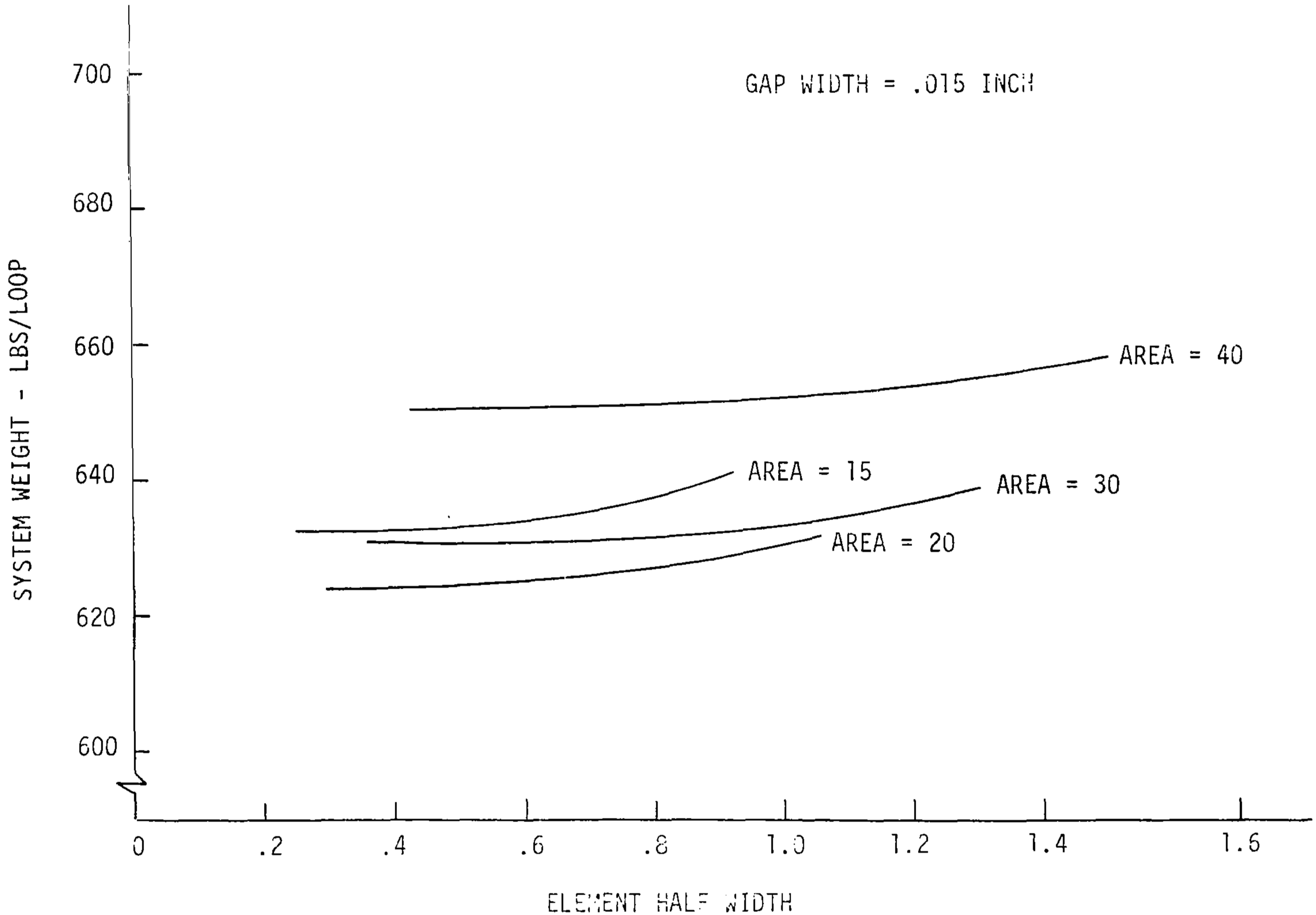


FIGURE 5-1. VARIATION OF LOOP WEIGHT WITH GAP HALF WIDTH AND WICK AREA

The first computer runs were made using approximated values for each combination of parameters. These approximated values were obtained by making engineering guesses at the reasonable heat transfer areas required to transfer the imposed heat load, while maintaining a reasonably small temperature drop between condensing potassium vapor and the pump working fluid. Then, further runs were made in which each of the pump design parameters were varied both higher and below this reasonable value. These calculated results were then plotted as system weight versus, for example, the element half-width with area as a parameter with a different plot for each gap width.

Figure 5-1 is an example of the data obtained. This figure is for a gap width of 15 mils for four pump loops employing potassium as a working fluid. The calculations showed that the system weight is practically independent of the element half-width for values up to about 1 foot, and for gap widths greater than 15 mils. Since it is not practical to build a pump system with a gap smaller than 15 mils, the element half-width was eliminated as an important variable.

The system weight becomes smaller with smaller gap widths. This trend is because the smaller the gap width results in a smaller temperature gradient and a higher working fluid temperature in the pump, and therefore, in the radiator. Radiator weight varies approximately as the inverse of the inlet temperature to the fourth power and is clearly the dominate weight in pumper system. The gap width should be the smallest which is reasonable to manufacture. Experience with heat pipe systems indicates that 15 mils is a reasonable lower limit.

The calculated results showed that the overall system weight also depends upon the heat transfer area in the pump. There is, in fact, an area for which the system weight becomes a minimum. Figure 5-2 illustrates the system weight versus the capillary pump area for a gap width of 15 mils. The system has four capillary pump loops using potassium as a working fluid a 12 inch (.305 m) separation distance between feed arteries. This figure shows that a minimum loop weight of 624 pounds (283 Kg) occurs at a heat transfer area of 22 ft^2 (2.04 m^2). (The total system of four loops is 2496 pounds (1132 Kg) and 88 ft^2 (8.18 m^2.)

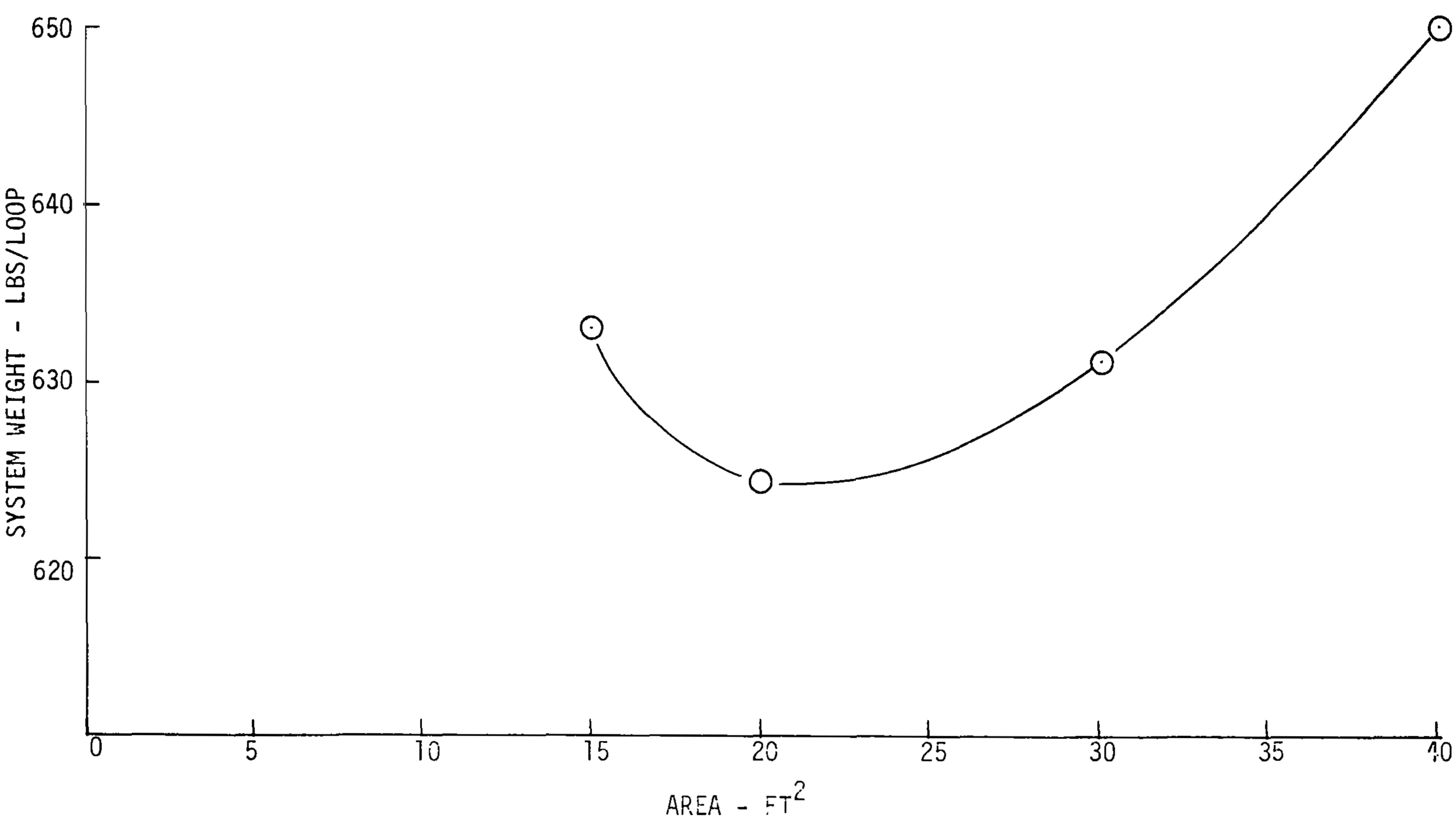


FIGURE 5-2. SYSTEM WEIGHT VERSUS WICK/AREA FOR GAP WIDTH = 0.015 INCH AND HALF WIDTH = 0.5 FEET

Similar results were obtained when calculations were made for numbers of loops of 8, 12, 16, and 20, and using other working fluids of sodium and cesium.

When the minimum weight systems for the various working fluids were compared at a constant system variable of loop numbers, it was found that for each of the numbers of loops, potassium resulted in the lowest overall system weight. This advantage with potassium was not overwhelming but was consistent. This can best be explained by noting the short-comings of the other two fluids; sodium and cesium. In the case of sodium, the vapor pressure at the maximum temperature of about 1200°F (922°K) is only about 1 psi (7000 N/m^2). Therefore, to maintain the pressure drops around the loops to be less than this 1 psi (7000 N/m^2) (and they must be considerably less), the dimensions of all the piping, etc., must become very large. On the other hand, in using cesium the surface energy of cesium is half that of potassium and one obtains only about half a psi (3500 N/m^2) pressure increase across the same screen. In order to maintain the pressure drops around the loop less than 1/2 psi (3500 N/m^2) requires that all the loop piping be very large, and consequently, the weight is large. An alternative is to use finer mesh screen. However, 400 mesh was considered the minimum to be handled practically and this alternative was discarded.

In summary, the capillary pump will employ a 400 mesh screen wick, spaced 15 mils from the hot wall. The wick will be fed potassium, the selected working fluid, from arteries spaced at about one foot (0.305 m), the exact spacing to be determined by the amount of area in each pump.

5.1.3 Selection of the Number of Loops

The selection of the number of capillary loops must be based upon the system weight, survivability requirements, controlability, manufacturability, reliability, and practicality. The latter three or four items are very difficult to quantitize in this application because the capillary pumped systems are still in a conceptual stage. These considerations are also strongly related to the performance and control characteristics of the total ARCPS. Experience gained through construction, test, and operation would provide better judgements about these factors. In the meantime,

engineering experience with related systems has been used, qualitatively, to make judgments relative to these factors.

The survival requirements of the system are that it has 75% of its heat rejection capacity at the end of its life, suggesting that the number of loops should be a multiple of four. Four loops allows one radiator penetration per lifetime, eight allows two, twelve allows three, etc. There is no advantage, for example, of nine, ten, or eleven loops at least from a weight standpoint, since all radiators must be armored for a total of only two penetrations. The use of these in-between numbers results in greater survival capability than required at the cost of system weight.

The design program was used to make weight trade-off studies. This was done by calculating, for each number of loops considered, the optimum weight system assuming that each loop removed an equal share of the total heat; i.e., the total heat load divided by the number of loops. The importance of this assumption is that the system will actually be larger than calculated because the last loops in the series will not remove as much as the first. The redesign of the system using the support of the performance program has shown that the actual loop design is about 15% larger than calculated using the average heat rejection. However, using $\frac{Q}{N}$ as the individual loop heat rejection provides a common basis for comparison to establish the optimum number of loops.

Figure 5-3 shows a plot of the total heat rejection system weight as a function of the number of loops. These data show that a broad minimum exists between about twelve and twenty-four loops. However, the change in system weight from the maximum to the minimum (about 10% change) is very small, probably smaller than the calculation accuracy. It seems clear that the system weight is not a major factor in determining the number of heat rejection loops at least for the range considered.

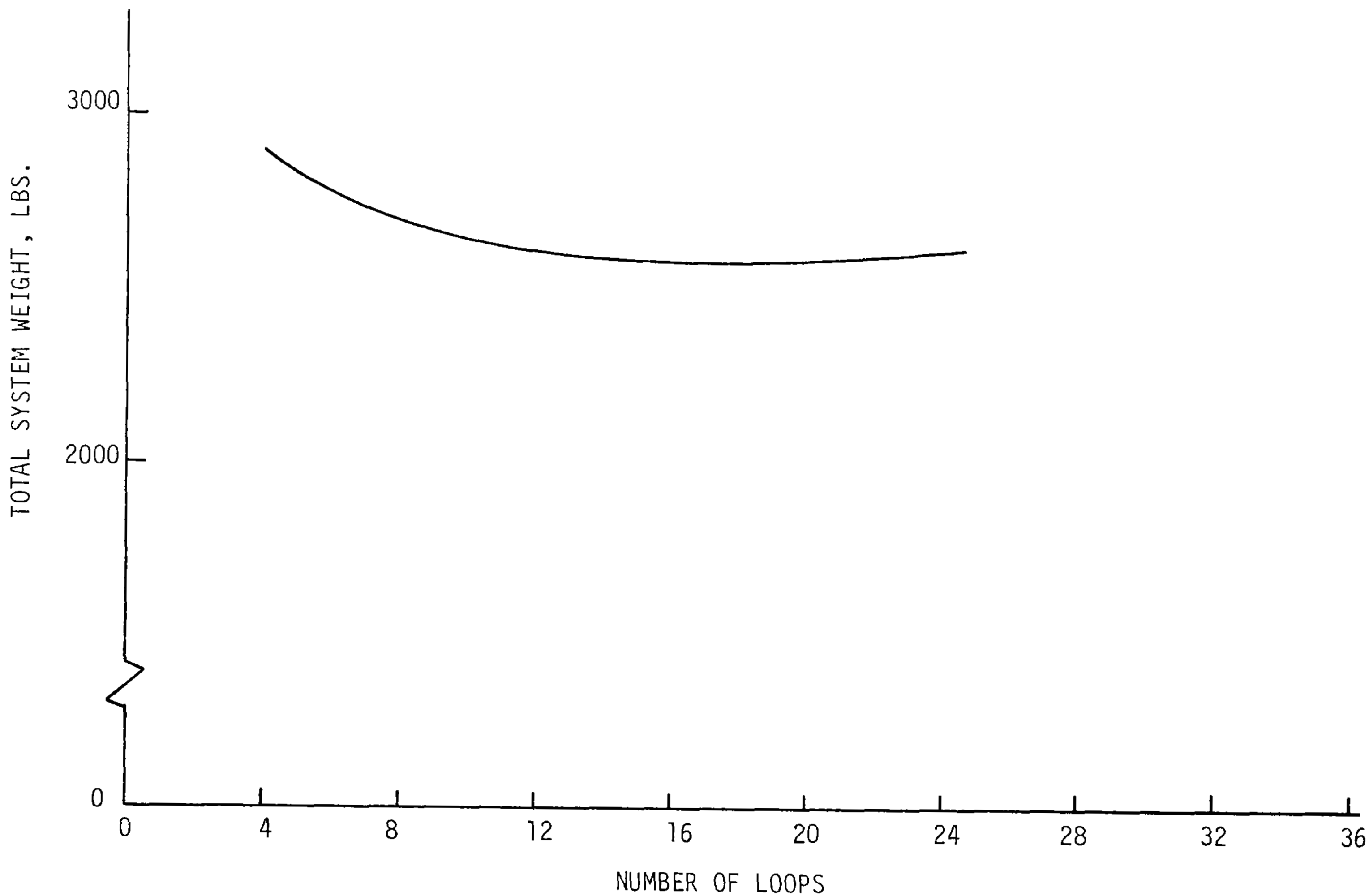


FIGURE 5-3. TOTAL HEAT REJECTION SYSTEM WEIGHT VERSUS NUMBER OF LOOPS

The frequency at which power adjustments must be made in the case of the failure of a loop must also be considered when choosing the optimum number of loops. The amount of heat to be rejected is divided into about 8% sensible and 92% latent. Thus, with twelve loops, the latent heat is removed in the first eleven loops and the last loop provides the subcooling. With fewer loops, the last loop removes latent and sensible energy. Consequently the failure of a loop would result in vapor leaving the condenser. When the sensible heat is removed by one or more loops, a radiator loss does not require an immediate power adjustment. When the sensible heat is removed by many loops, say ten of a 120-loop system, a readjustment of the power may not be needed at each loop failure since the extra rejection capability of all the loops may be enough to make up for the loss of a single loop. The only effect on the condensing potassium side will be a slight increase in the outlet temperature. In this sense then it is desirable to have a larger number of loops since it allows for use of a simple low response time control system. It is, in fact, possible to rely on operator or ground compensation of a loop rather than automatic control. Twenty-four loops would be approximately the minimum number to allow for a low response, possibly remote, control system.

The radiator configuration selected has 120 parallel tubes which represents a practical upper limit from a packaging standpoint. Thus, 120 loops would have a separate capillary pump for each tube in the radiator. About ten of these pumps would remove sensible heat; i.e., would subcool the potassium.

The weight of a 120-loop system was estimated to be approximately 10% greater than the 24-loop case. However, this estimate does not include auxiliary equipment associated with each loop (sensors, controls, mounting hardware, etc.) whose total weight is roughly proportional to the number of loops. Consequently, the total system weight of a 120-loop case should be substantially more than the 24-loop case. One hundred twenty loops represents more fabrication problems and therefore potentially represents a less reliable system. However, this facet of the evaluation cannot be quantitized at this stage of the system development.

It should be noted that the selection of an acceptable number of pumper loops, since it does not severely impact the overall system weight, allows a broad choice and can be tailored to meet the geometry/packaging requirements of the particular system. This property is a very attractive attribute of capillary-pumped loops.

Most of the emphasis in the performance studies were made using 12 and 24-loop capillary pumped heat rejection systems. It is clear that this choice could not be completely validated by firm quantitative trade-offs and that other numbers (30, 60, 120) should not be discarded in future studies.

5.1.4 Radiator Optimization and Design

The radiator design program described in Appendix E was used to generate curves giving minimum weight radiators as a function of system operating conditions. These curves were then input to the design program to arrive at optimum weight systems.

Parameters affecting radiator weight can be divided into two types; internal parameters, those which do not impact the remainder of the system, and external parameters, those which affect the operation of other systems components. These parameters are summarized below:

Internal Parameters

- Number of tubes
- Tube diameter

External Parameters

- Two phase pressure drop
- Fluid type
- Number of loops
- Inlet temperature
- Amount of sub-cooling

The optimization procedure was to vary the internal parameter values for each set of external parameters until a minimum weight radiator design was found. In all runs, the subcooling was fixed at 100°F (56°K).

An example of the initial radiator optimization runs is shown in Figure 5-4. Primary purpose of these runs was to aid in determining the best fluid and to determine the sensitivity of radiator weight to the external parameters.

The external parameters which influenced the radiator weights most significantly were the fluid type, the number of loops (allowable meteoroid penetrations) and the inlet temperature. The difference between using cesium and potassium is very small, but the use of sodium results in significantly heavier radiators. This is because the sodium radiator tubes are larger in order to reduce the pressure drop.

The influence of allowable penetrations is determined by the tube armoring, the fewer penetrations, the heavier the armoring. The inlet temperature is important because the required heat transfer area is proportional to the fourth power of temperature.

The two phase pressure drop was not influencial except at very small values.

Curves of optimum raditor weights were used to make a series of parametric runs with the design program. The maps of optimum radiators for potassium given in Figures 5-5, 5-6, and 5-7 were then generated to further refine optimum system design and operating conditions. The final radiator design is given in Section 5.2.

5.2 SYSTEM CONCEPTUAL DESIGN

Préliminary designs of 12, 24, and 120 loop heat rejection systems were made. This section describes these results.

The design calculations reported in Section 5.1 established several design parameters. These are the following:

- o Potassium is the working fluid.
- o The wick is one layer of 400 mesh screen. (For reliability it may be desirable to use at least two layers. This was not, however, implemented in this design study.)
- o The spacing between the condenser wall and the screen wick is 15 mils.
- o The spacing between feed arteries is approximately 12 inches (.305 m).

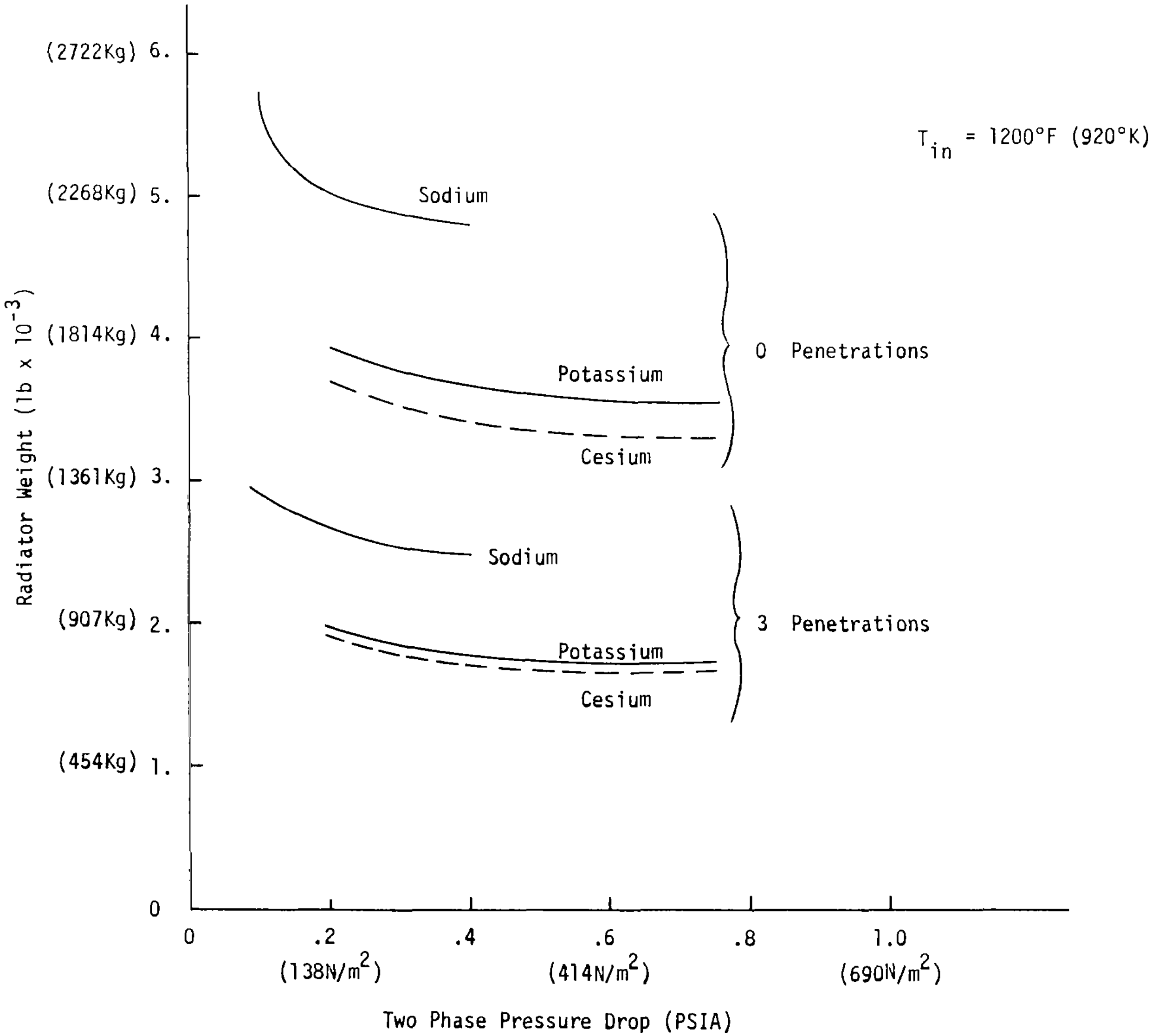


FIGURE 5-4. OPTIMUM RADIATOR WEIGHT COMPARISON

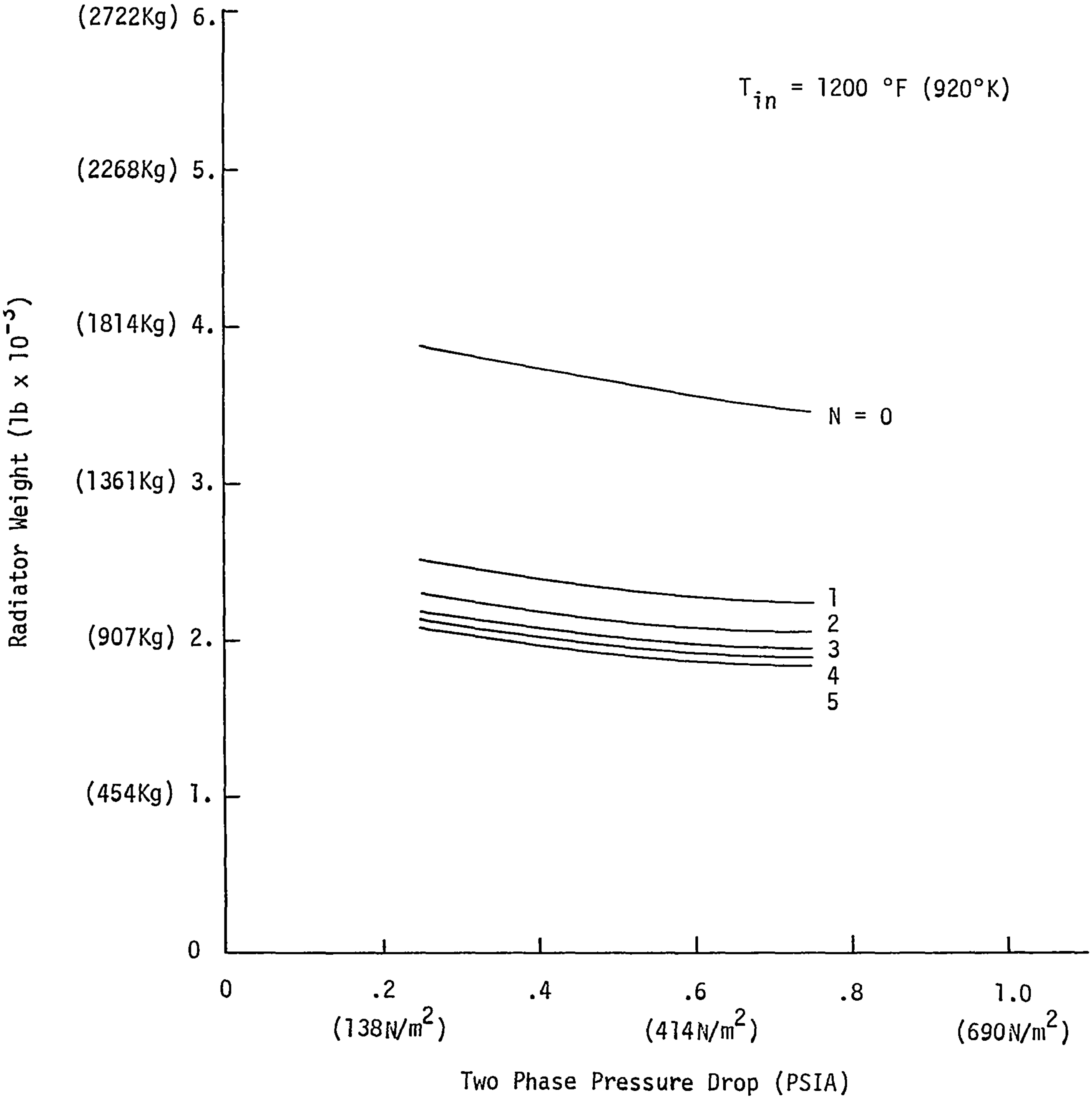


FIGURE 5-5. OPTIMUM POTASSIUM RADIATOR WEIGHTS

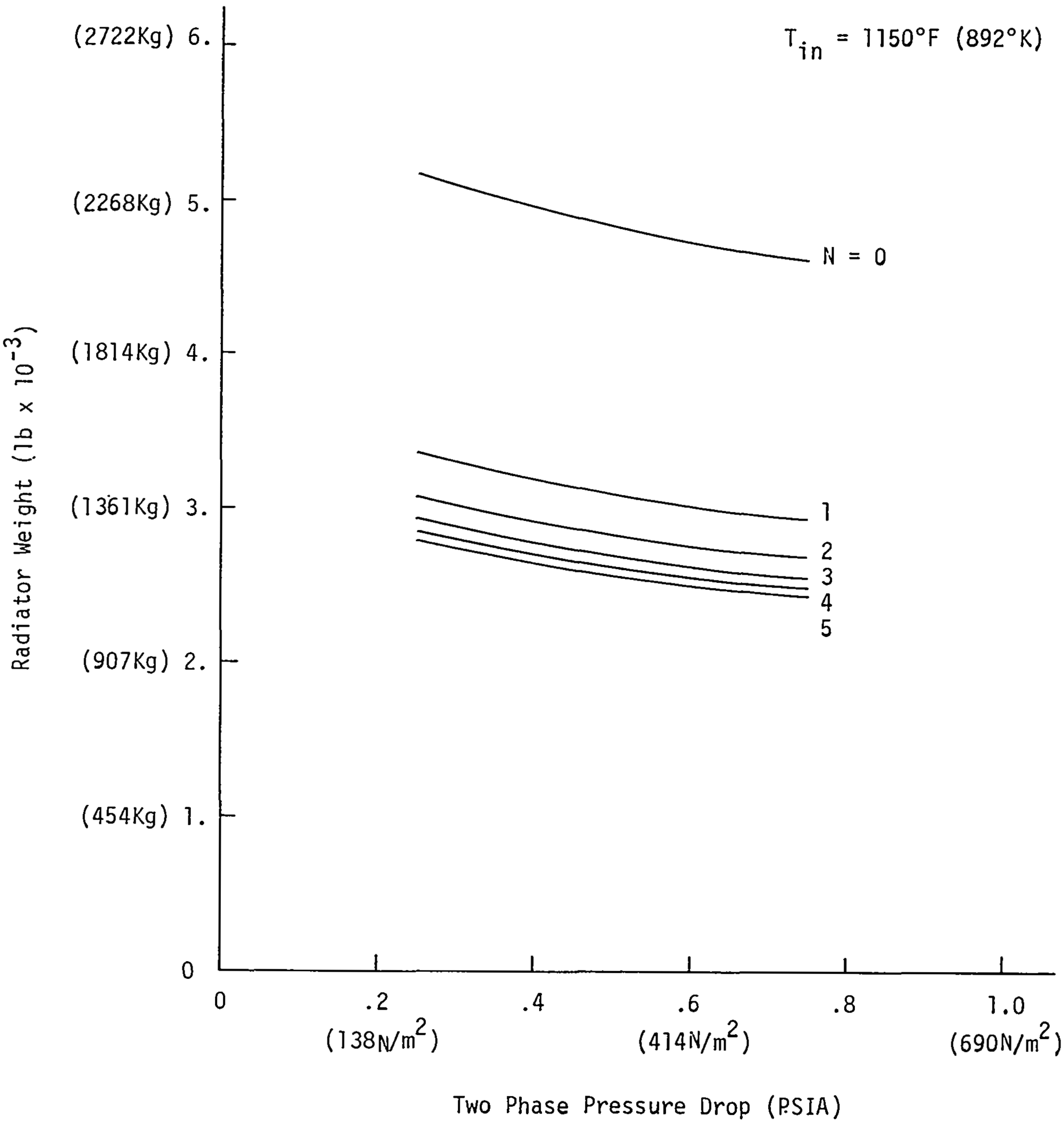


FIGURE 5-6. OPTIMUM POTASSIUM RADIATOR WEIGHTS

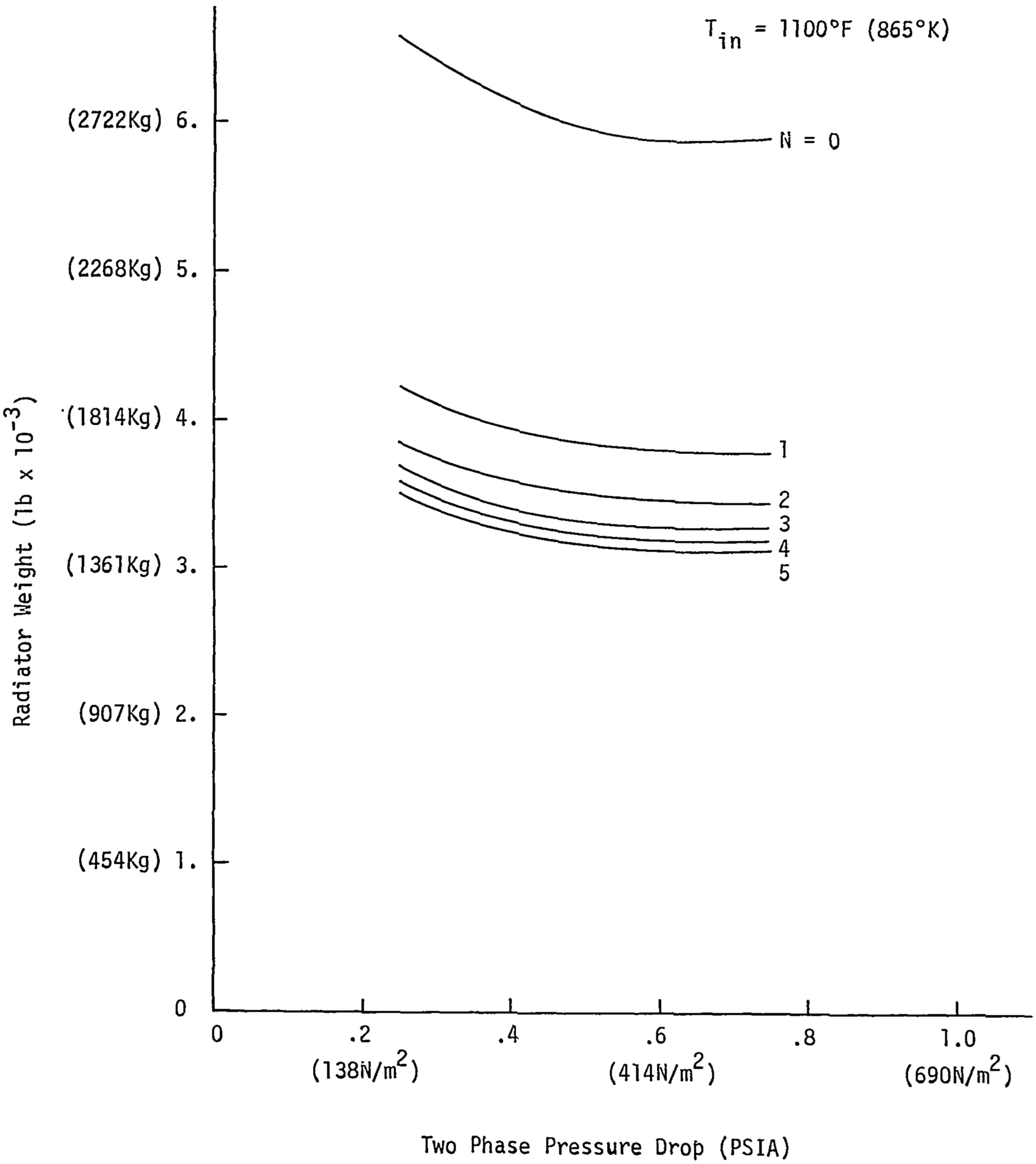


FIGURE 5-7. OPTIMUM POTASSIUM RADIATOR WEIGHTS

In addition, the approximate minimum weight dimensions of the capillary pump and the radiator were determined using an average heat rejection requirement (total heat load/number of loops). Actually, the temperature of the condensing potassium stream decreases as it flows past each pump in the series, and therefore, each loop removes a different amount of heat. The individual loop must be designed to remove more than the average amount of heat in order to provide adequate total capability.

The design program and the performance program were used in an iterative fashion to determine the actual pump and radiator size required to reject the total waste heat load. The iterative procedure was to use the design program to design an optimum loop which removed (1+n) times the average where n is a variable fraction. Then the series of loops so designed were evaluated by the performance program to determine if the system would remove the imposed heat load. If the total system removed too little heat, n was increased. If it removed too much heat, n was decreased.

The dimensions of the capillary pump designed in this way are summarized in Table 5-5 for a 12-loop configuration. Figure 5-8 gives a scaled drawing of a pump which furnishes the necessary heat transfer area and which fits into the geometry of the truncated cone of Figure 2-3. The pump forms a hexagonal shape which "circles" the top of the truncated cone as shown by the top view of Figure 5-9.

Table 5-5. Capillary Pump Dimensions

Heat Transfer Area	=	6.4 ft^2 (0.595 m^2)
Gap Width	=	0.015 inch (0.00038 m)
Liquid Line ID	=	0.77 inch (0.0196 m)
Liquid Line Length	=	25 feet (7.62 m)
Vapor Line ID	=	2.4 inch (0.061 m)
Vapor Line Length	=	5 feet (1.521 m)
Artery ID	=	0.65 inch (0.0165 m)
Artery Length	=	30 inches (0.762 m)
Vapor Space Dimensions:		
Height	=	30 inches (0.762 m)
Length	=	33 inches (0.838 m)
Width	=	0.8 inch (0.0203 m)

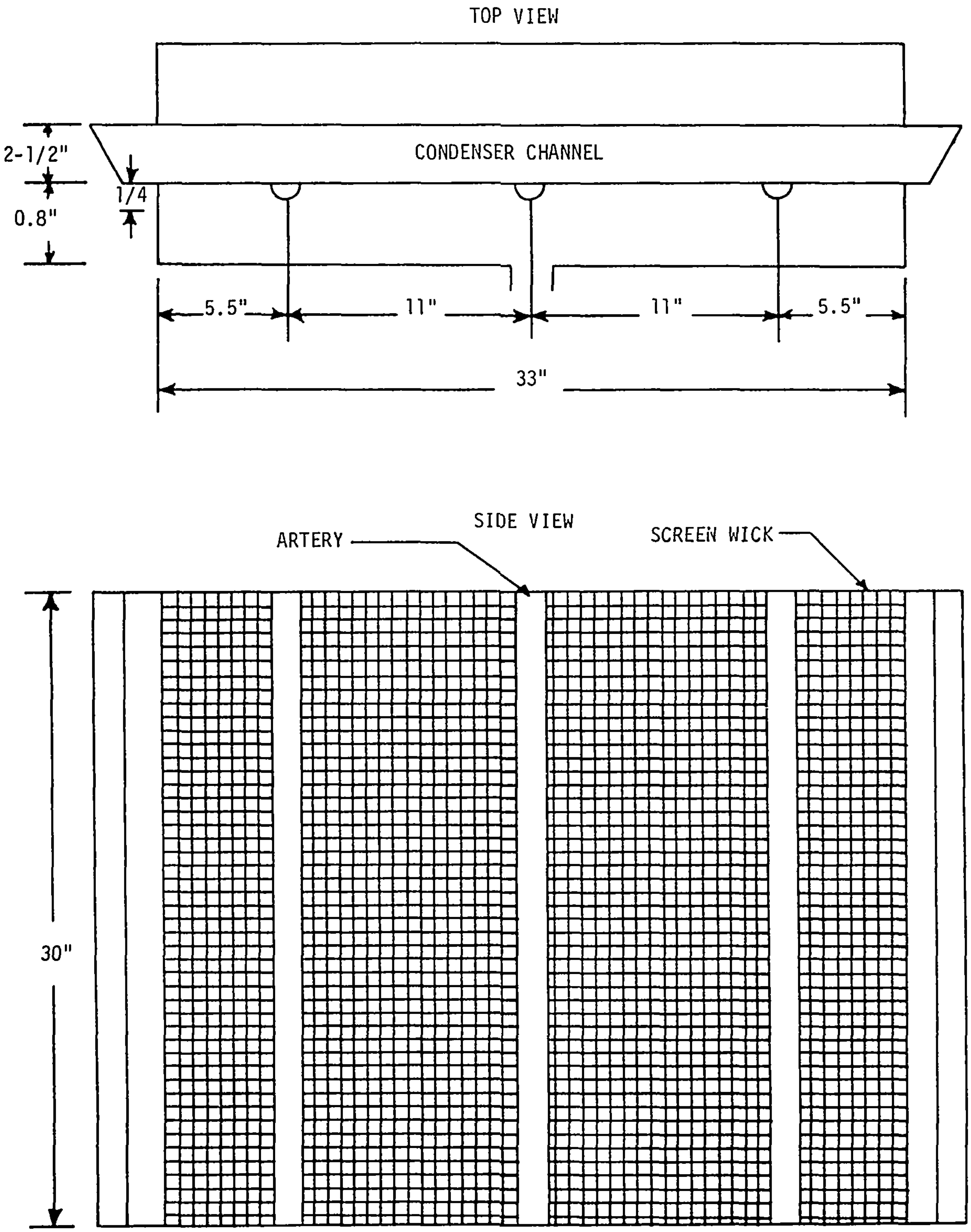


FIGURE 5-8. SCHEMATIC OF 12 LOOP CAPILLARY PUMP

The liquid feed arteries are hemi-cylinders of 0.65 inches (0.0165 m) in diameter. They were sized so that the combined liquid flow area of all the arteries would be the same as the liquid line from the radiator. The number of arteries was made to be equal to the smallest number that could be fitted into the pump with the restriction that no artery would be required to feed more screen than a strip one foot wide (0.305 m).

The vapor tube connects to the pump at the center of the screen area. This way the vapor generated flows radially inward converging on the exit. The depth of the pump was determined so that the maximum velocity, at the circumference of the exit, was equal to the vapor velocity in the vapor line.

The liquid line and the vapor line were sized so that the liquid velocity was about one foot per second (0.305 m/sec.) and so that the vapor Mach number equal about one-fourth. These velocities have been found to be realistic without excessive pressure losses (14). The lengths of these lines were estimated at effective lengths of 25 and 5 feet (7.62 and 1.52 m), respectively. The actual physical length will be smaller by 20 to 25%.

The radiator which is coupled with this capillary pumped system is shown in Figure 5-10. It fits on the outer surface and extends down the length of the truncated cone 18.3 feet (5.58 m). The complete radiator has 120 parallel tubes of 1.15 inches I.D. (0.029 m). Thus, each capillary pump feeds vapor to ten of these tubes. The vapor enters the tubes through a common header at the top, and the condensate is collected at the bottom of the tubes, again, in a common header and transported back to the pump.

The heat rejection systems employing 24 and 120 loops were not obtained using the iterative application of the design and performance programs as described above. Since the overall weight trade-offs indicated no significant effect of number of loops, these additional designs could be obtained simply by scaling down the capillary pump and the radiator proportionally to the number of loops. Thus, the screen area of a 24-loop capillary pump is 3.2 ft^2 (0.297 m^2) and for a 120 loop pump 0.64 ft^2 (0.0595 m^2). The 24 loop system used five radiator tubes for each pump and the 120 loop system used just one tube per pump.

Figures 5-11 and 5-12 show a top view of the 24 loop and the 120 loop systems, respectively, coupled to the radiator.

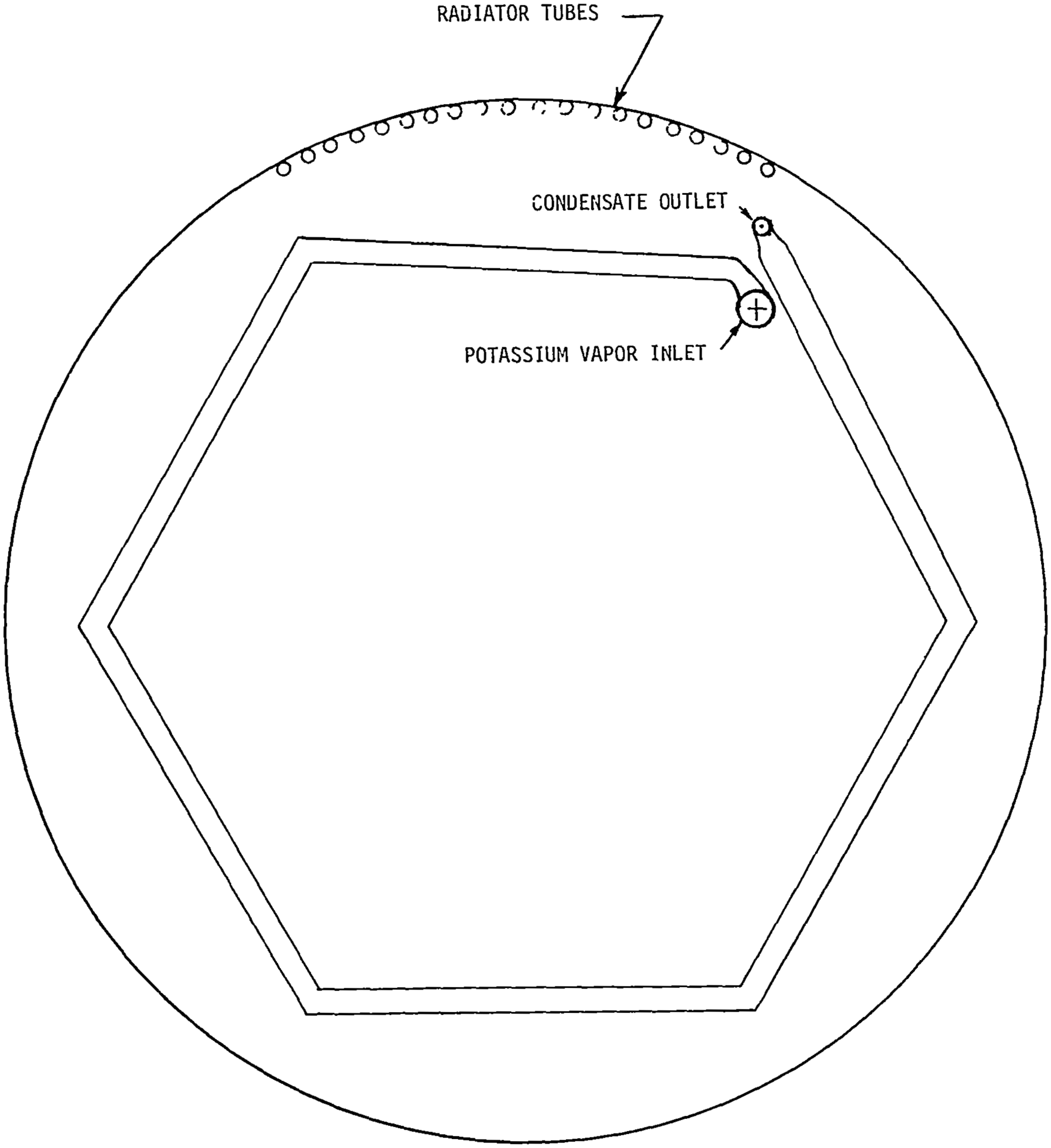


FIGURE 5-9. HEAT REJECTION SYSTEM LOCATION

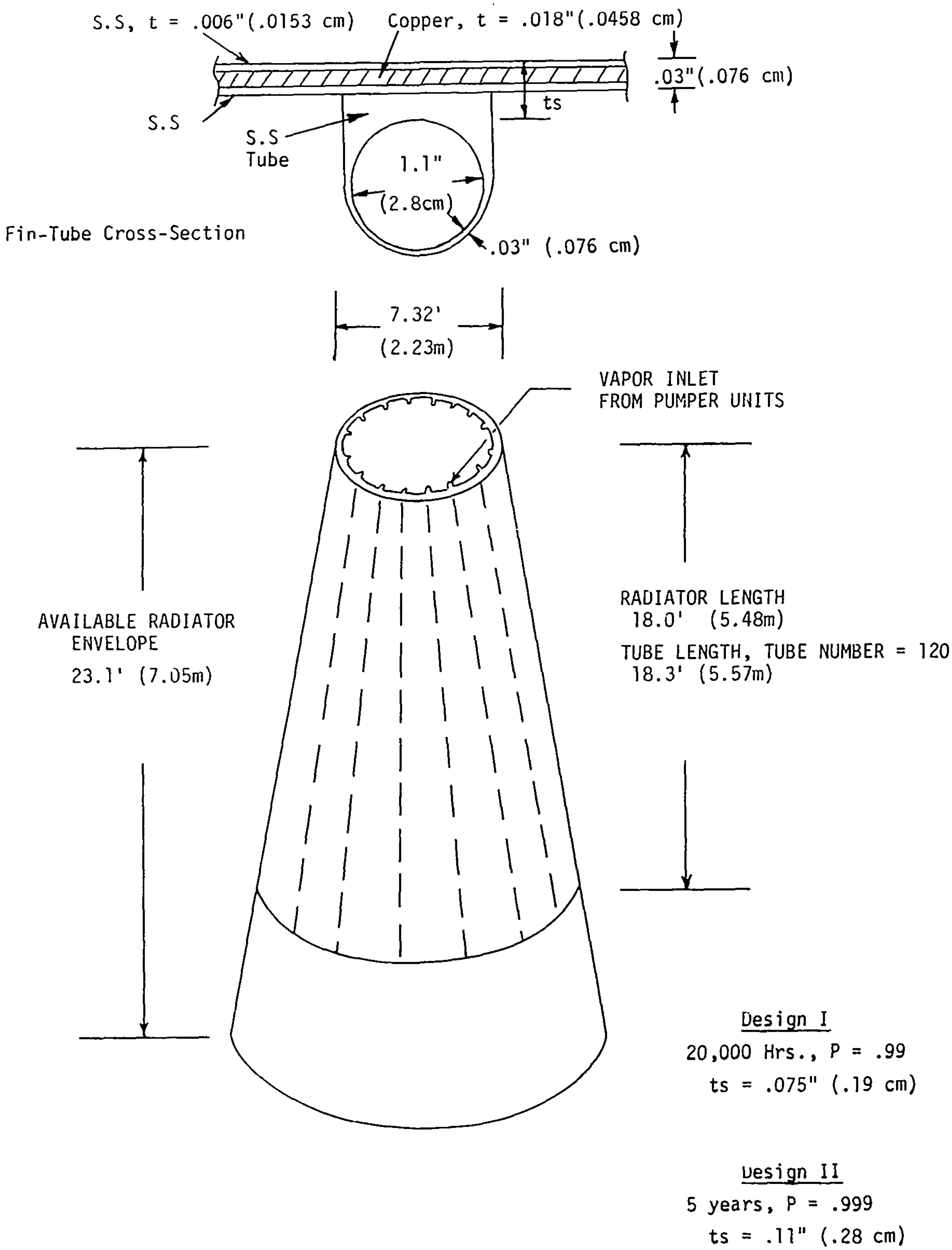


FIGURE 5-10. RADIATOR DESIGN

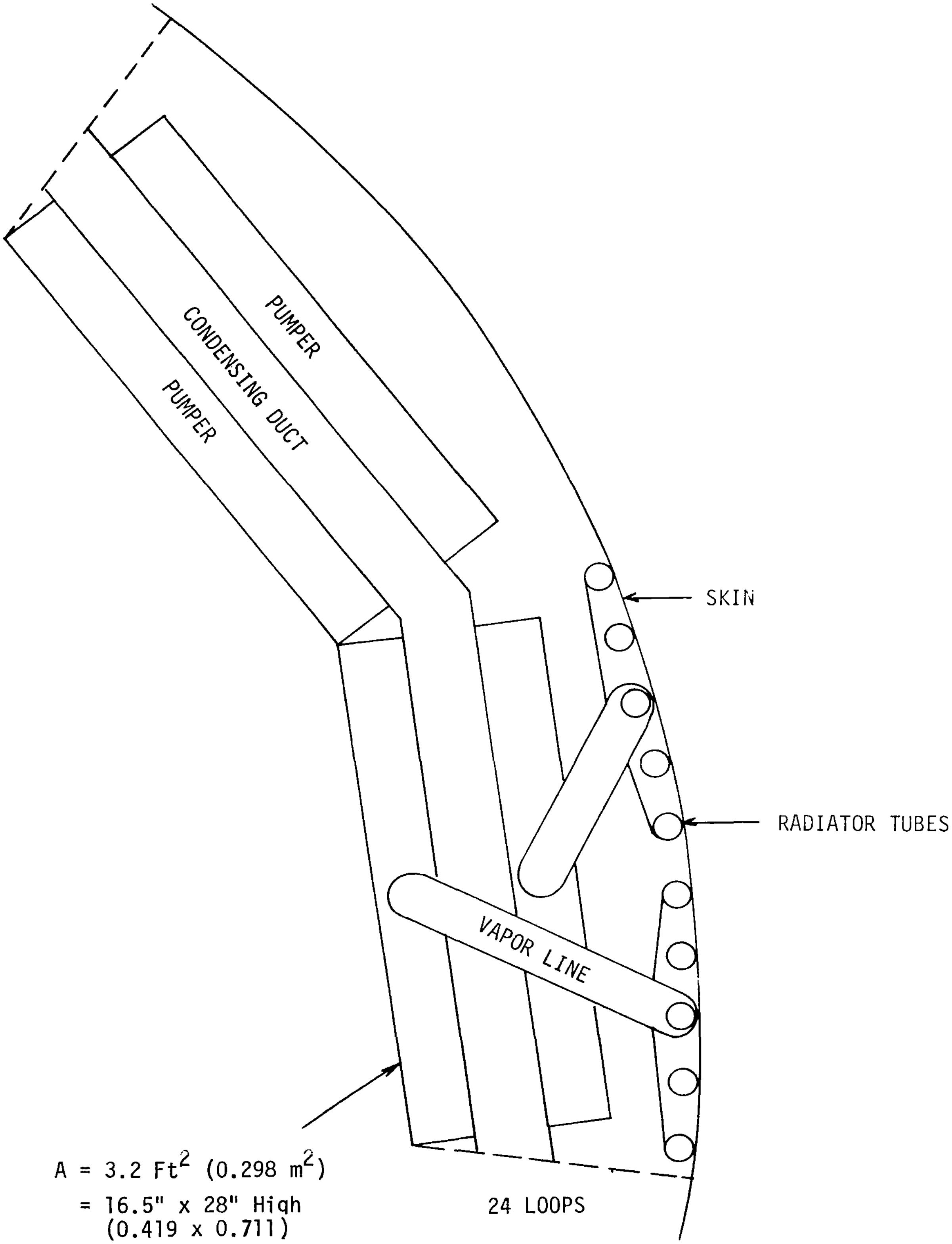


FIGURE 5-11. TWENTY-FOUR LOOP SYSTEM - RADIATOR COUPLING

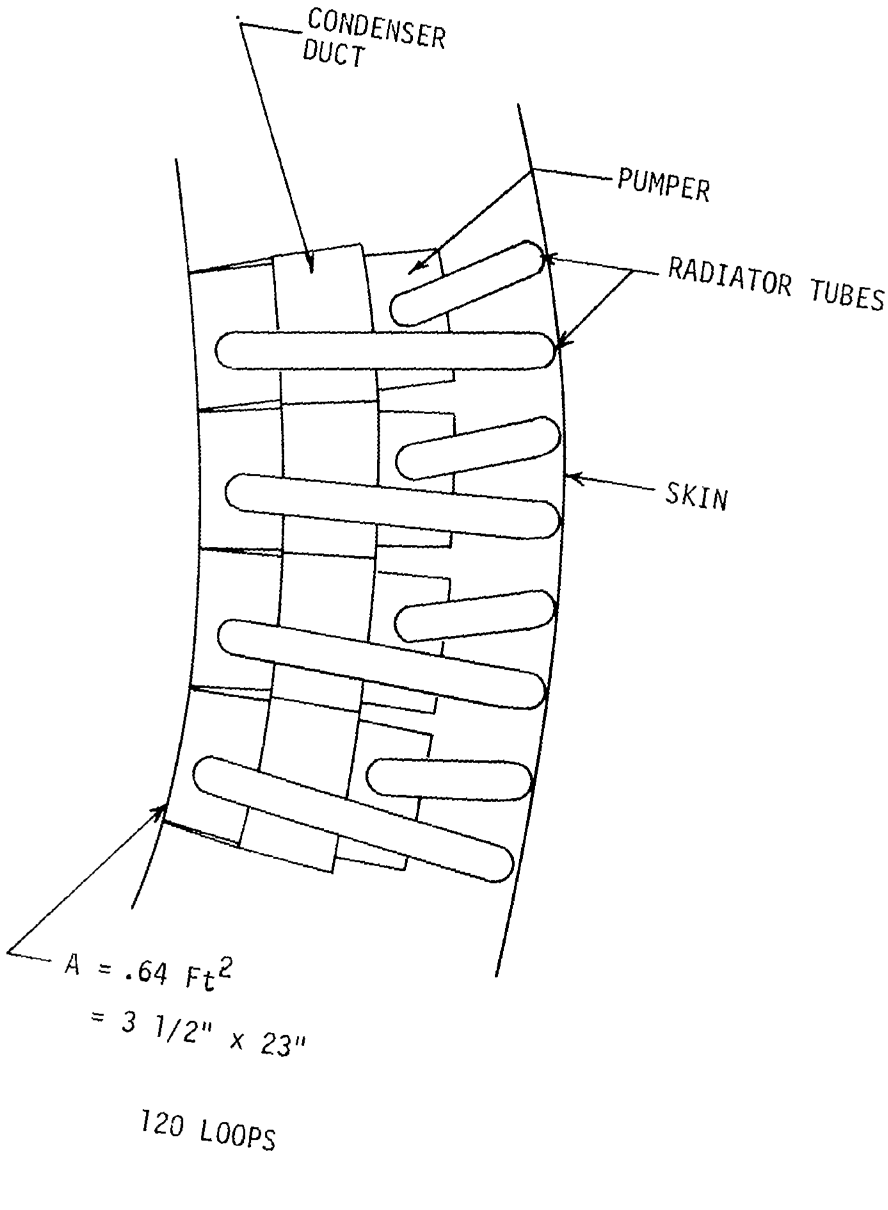


FIGURE 5-12. 120 LOOP SYSTEM - RADIATOR COUPLING

5.3 SYSTEM PERFORMANCE

Performance calculations were made to determine the normal operating characteristics for a twelve and a twenty-four loop heat rejection system. Additional performance calculations were made to determine the system steady state response to a single or multiple radiator failure. The latter calculations were made only for the twelve loop system. However, the physical mechanism of the response of this system is consistent and the expected response of systems with a different number of loops can be projected from the twelve loop results. This section of the report describes the results of these performance calculations.

5.3.1 Normal Operating Characteristics

The normal operating characteristics of the heat rejection system are given as profiles of the relavent operating parameters through the system as heat is removed. The sensitivity of these system performance profiles to pertubations in the design conditions and system dimensional parameters are, in addition, important characteristics of the system.

The profiles of the condenser temperature and pressure for a 12 loop system are shown in Figure 5-13. The corresponding pumper temperature and pressure profiles are shown in Figure 5-14.

Figure 5-13 shows the condenser side temperature decreasing as the flow passes through the condenser channel. The program assumes that in the quality region the temperature is the saturation temperature at the corresponding fluid pressure. This neglects the temperature drop across the liquid film at the wall, however, this is likely to be small. Consequently, as the pressure drops, due to the two-phase friction, the corresponding temperature drops simultaneously. This temperature drop as noted below, proves to be a very important factor in the design of the system.

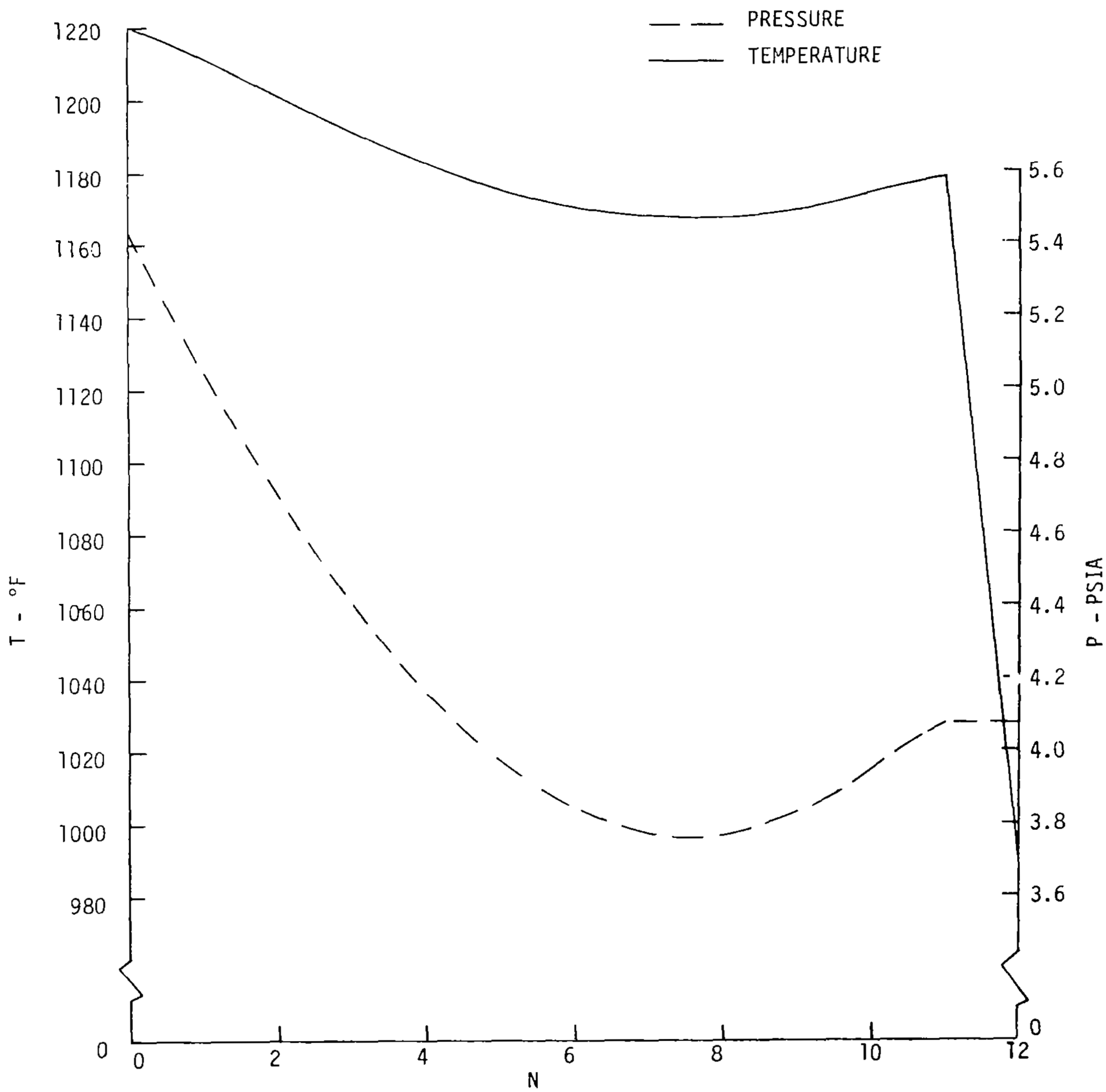


FIGURE 5-13. NORMAL CONDENSER PROFILES FOR 12 LOOP SYSTEM

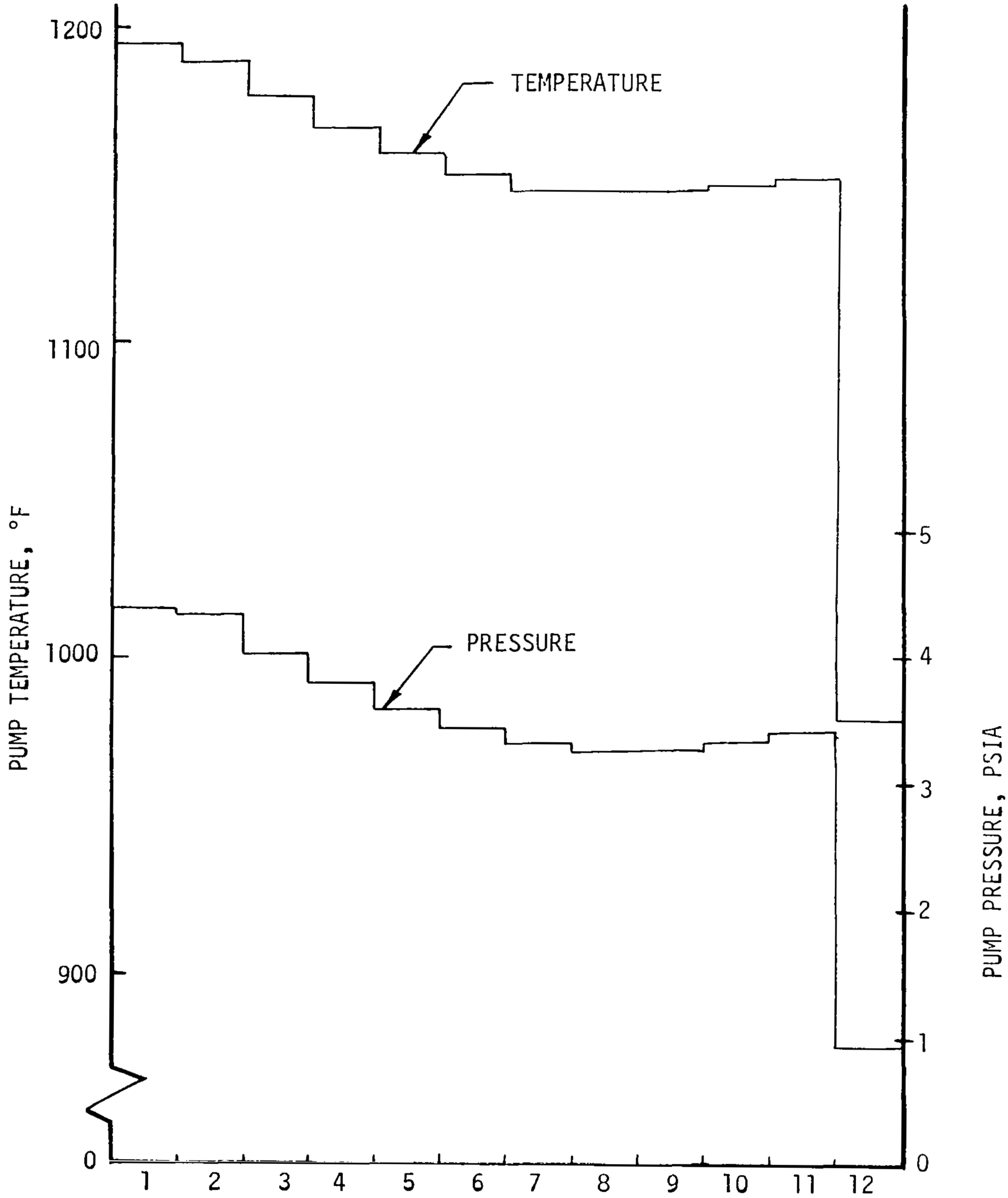


FIGURE 5-14. CAPILLARY PUMP TEMPERATURE AND PRESSURE PROFILES

The condenser side pressure drop is composed of two components. There is a two-phase frictional component which always decreases in the direction of flow, and the second is a deceleration pressure rise brought about by the decreasing volumetric flow rate. The deceleration pressure rise is fairly constant throughout the channel since the rate of quality decrease is essentially linear. Near the entrance to the condenser channel, the frictional pressure drop is relatively high due to the high velocity of the stream. As the potassium stream passes each pumper, the quality decreases and therefore, the magnitude of the frictional pressure drop decreases. At approximately pump number eight the frictional pressure drop equals the deceleration pressure rise and the net pressure change is zero. For the following three pumps, the frictional pressure drops become smaller than the deceleration pressure rise and the total pressure increases slowly until the quality equals zero. Thereafter, the deceleration pressure change is zero and the frictional pressure drop becomes extremely small.

The static pressure profile represents a potential stability problem. As the pressure drop approaches zero and begins to become positive, the stream tends to flow backwards creating large turbulent eddies. This enhances the flow void feedback character in the power conversion loop to produce flow oscillations. These stability problems are discussed in more detail later in this report.

The use of a tapered condenser passage to give a continuously decreasing pressure was given consideration. However, the performance calculations showed that when a radiator failure occurred and a fraction of the heat was no longer removed, two-phase flow extends further down the tube into an area of smaller cross-section. The results of this are a very high pressure (and corresponding temperature) drop in the potassium and a reduction in the overall heat transfer rate from the system. This process degenerates the operation of the system. It is clear that tapering of the condenser tube would require a very close control system on the power conversion loop so, that simultaneously with a radiator loss, the power level can be decreased to maintain essentially the same pressure and temperature profile in the condenser.

The condenser side temperature profile was found to be one of the most influential parameters of the system. The optimum designed series heat rejection loop system was found to be able to remove more than required or far less than required, depending upon the shape of this profile. If the pressure drop in the condenser side became large, the temperature of the last pumper became too low for effective operation. The conclusion of these results is that the heat rejection system cannot be designed completely independent of the power conversion loop. These two loops must be designed simultaneously so that the pressure drop in the power conversion loop condenser is continuously decreasing, and simultaneously the temperature never becomes so low to allow effective pump operation.

One further note. Had sodium been the working fluid, this problem would have been more severe. On the other hand, with cesium it would have been less severe.

Figure 5-15 shows the quality profile through the condenser channel. The quality is very nearly linear with distance down the condenser tube and becomes equal to zero at the exit of the eleventh pump.

5.3.2 Performance with the Radiator Failure

The heat rejection system has been designed to permit some meteoroid penetrations with subsequent loss in radiators during its life. It is essential, then, to assess the change in steady state operating characteristics following such a loss.

The performance program was used to analyze this situation under two conditions. First, it was assumed that a radiator was lost but that no change in the condenser side heat load requirements was made. Then, a similar analysis was made in which the heat rejection requirements were decreased proportional to the fraction of the heat rejection capability lost. That is, when one radiator of a series of twelve is lost, the heat load is decreased to 11/12 of its normal value.

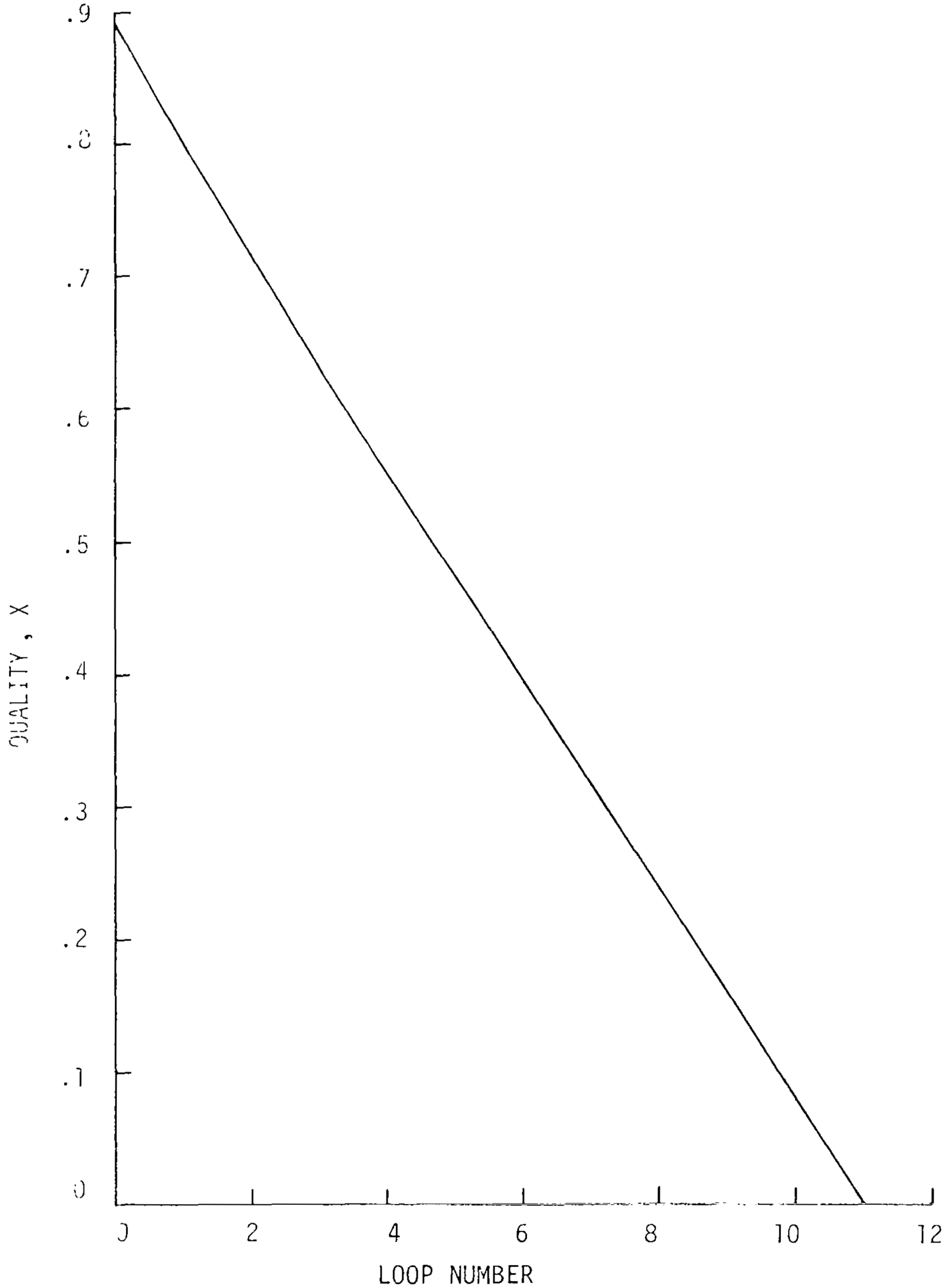


FIGURE 5-15. QUALITY PROFILE IN CONDENSER

In general, the result of this analysis was fairly simple. As any one radiator failed, and the corresponding pump became inactive, the effective pump numbering shifted by one. As an example, should pump number one be lost, then entrance conditions to pump one shifted to pump two with pressure loss and temperature and quality increase due to friction. In this way pump number two was essentially shifted to position one with a small change in inlet conditions.

The location of the pump lost makes some difference in the impact on the overall system. The further upstream this loss occurrs, the greater the influence. That is, in a twelve loop system the loss of pump number one had a more severe effect than the loss of pump number twelve. The explanation for this is simply that when pump number one is lost, the average operating temperature of the total system suffers a greater reduction than when the last is lost. As just stated in the previous paragraph, one loses essentially the same heat transfer capability as exists for that pump under normal operating conditions.

Figures 5-16 and 5-17 show the temperature and quality profiles in the potassium condenser tube as a result of losing pump number one with no change in the heat rejection requirement. Figures 5-18 and 5-19 show the corresponding temperature profiles as a result of losing pump number twelve with no change in the heat rejection requirement. Comparison of the quality profiles show, as discussed above, that the loss of pump one is more severe (higher potassium vapor quality leaving the condenser) than the loss of pump twelve.

Figures 5-20 and 5-21 show the temperature and quality profiles for a twelve loop system in which pump number one has been lost and the heat rejection requirement has been reduced to 11/12 of normal. These curves are to be compared with Figures 5-16 and 5-17.

It should be noted that the loss of any loop in a twelve loop system would have to be recognized quickly since it could lead to cavitation vapor lock of the boiler feed pump in the ARCPS. From this standpoint a twenty-four (or more) loop case would be advantageous since response to a single radiator failure could be tolerated or corrected at a more liesurely pace.

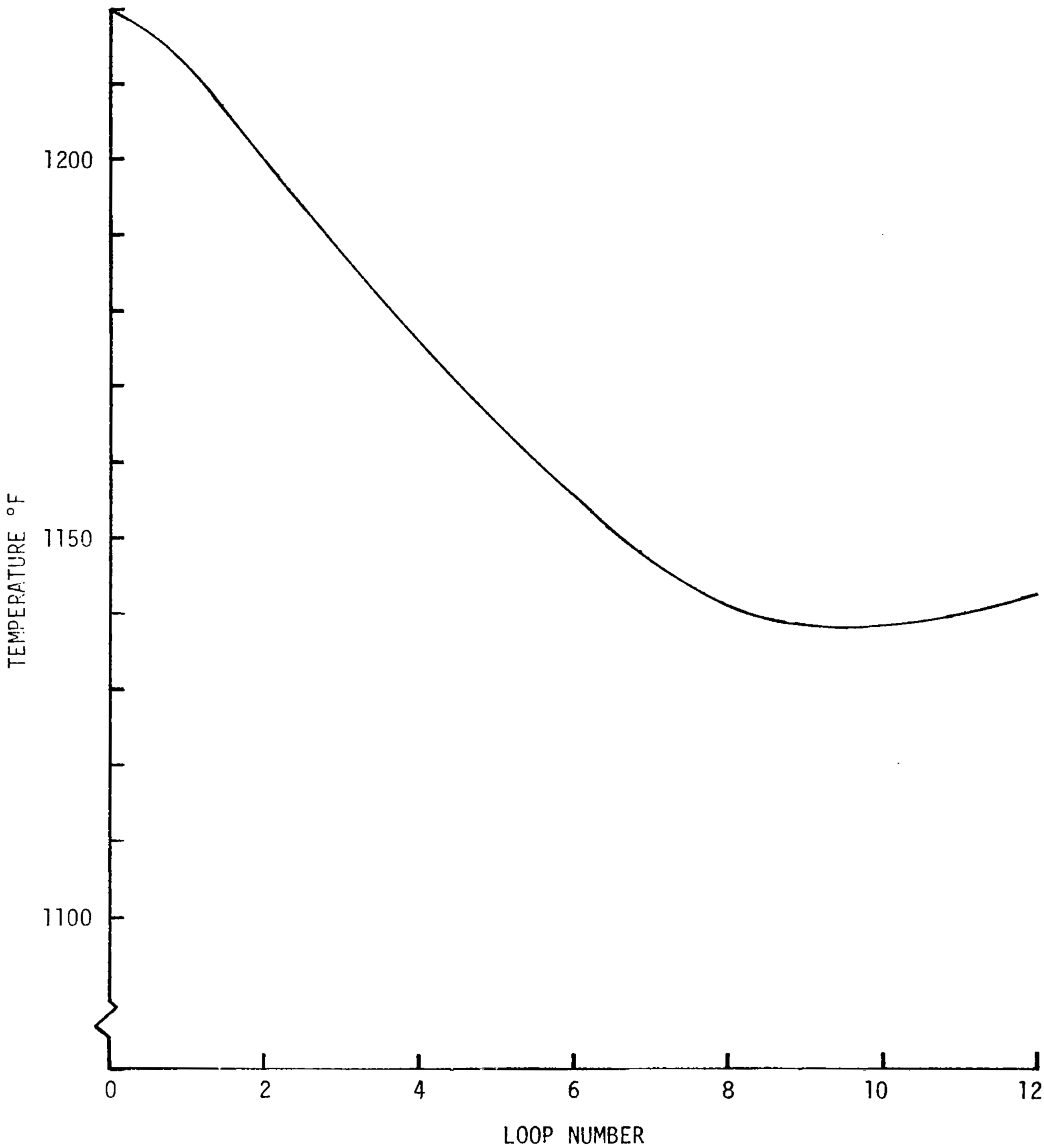


FIGURE 5-16 TEMPERATURE PROFILE WITH PUMP ONE LOST

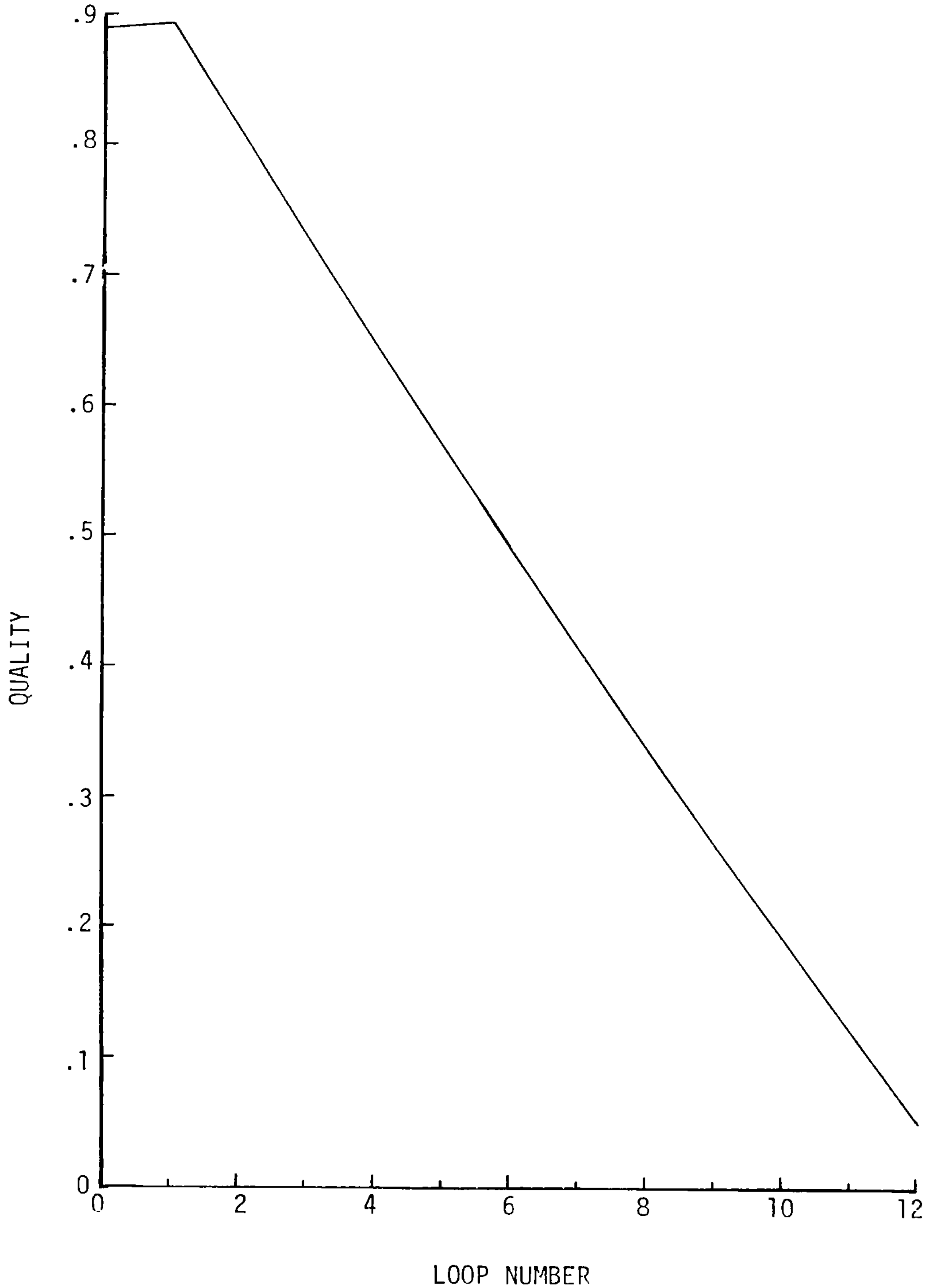


FIGURE 5-17 QUALITY PROFILE WITH PUMP ONE LOST

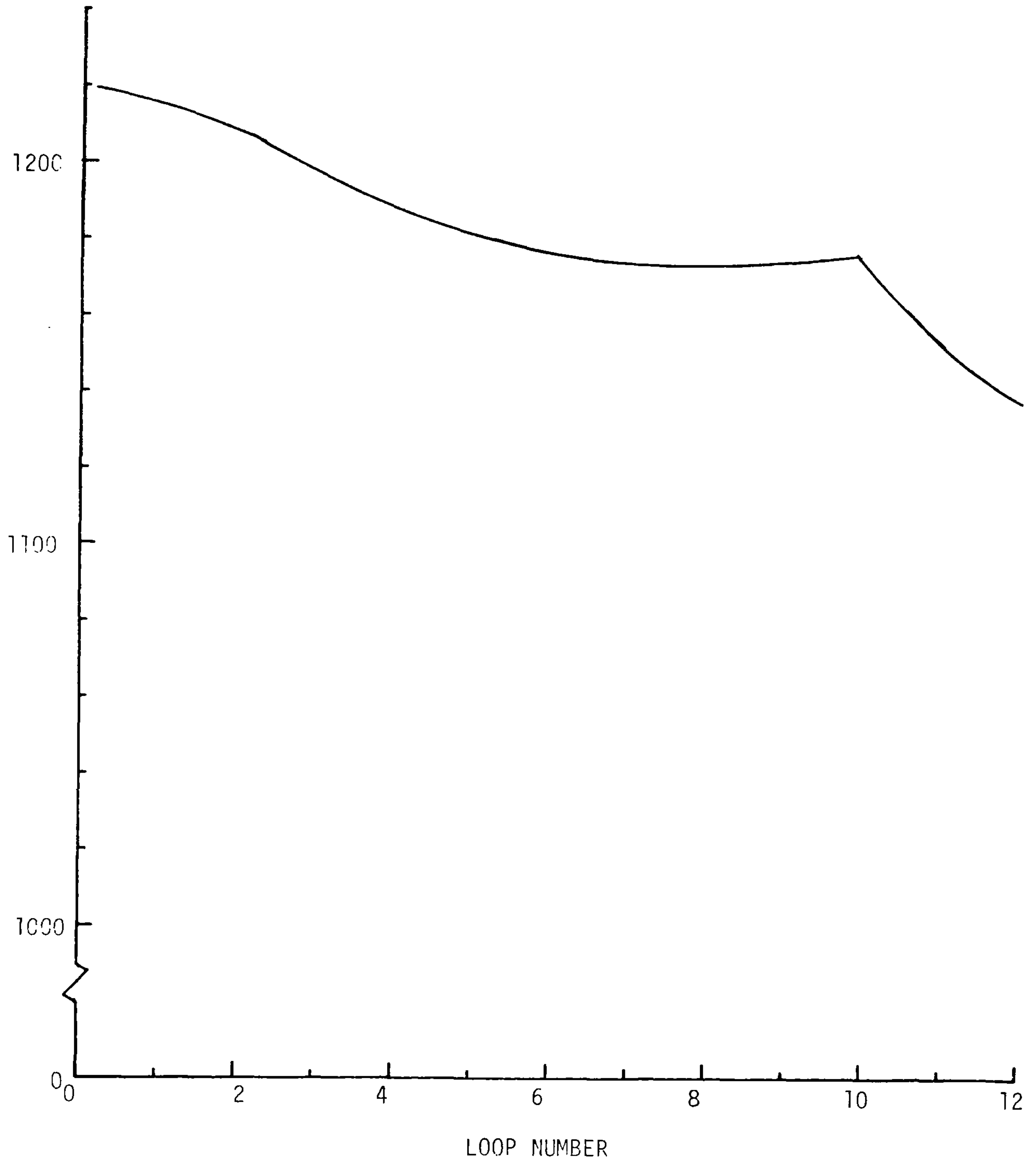


FIGURE 5-18 TEMPERATURE PROFILE WITH PUMP TWELVE LOST

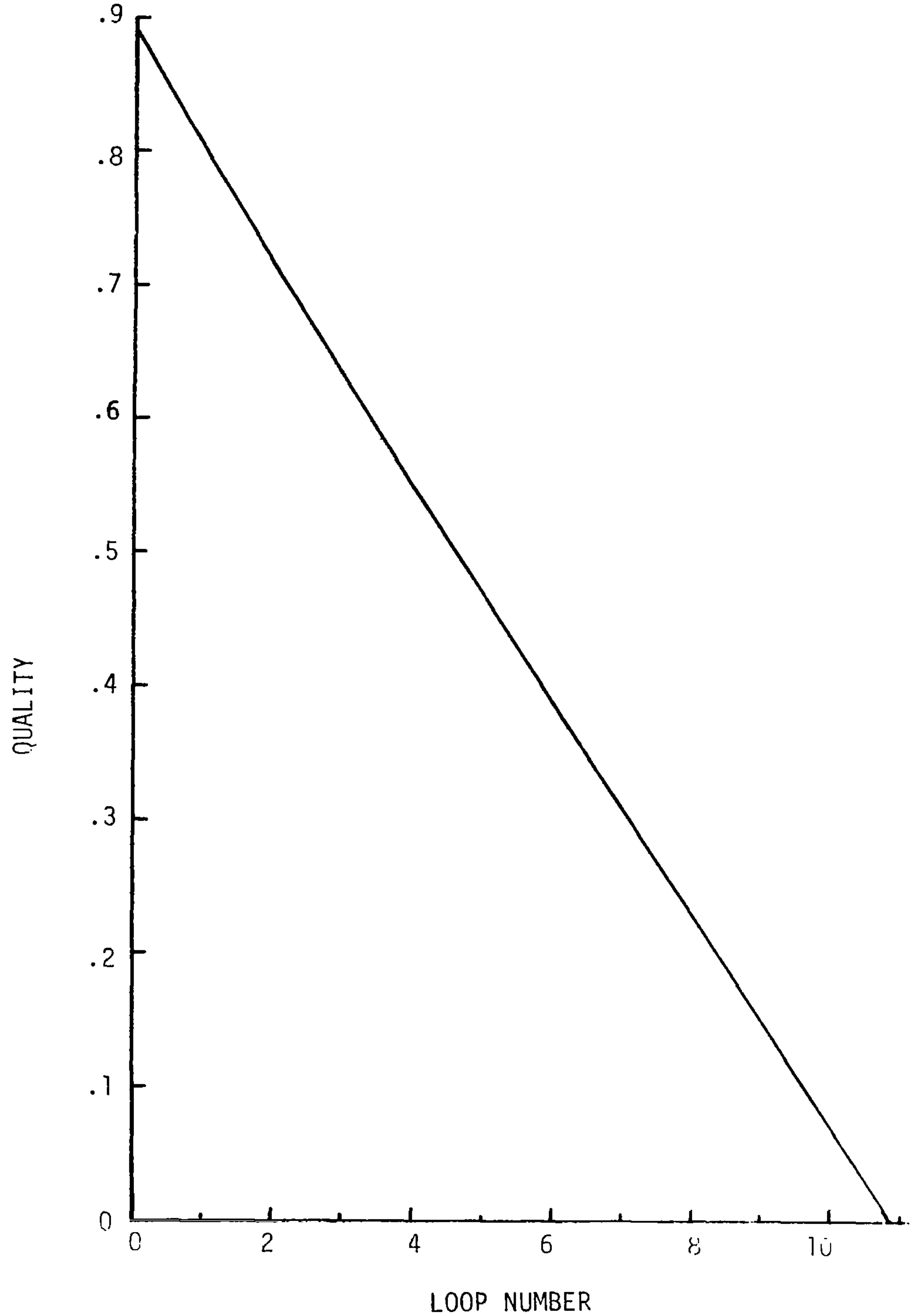


FIGURE 5-19 QUALITY PROFILE WITH PUMP TWELVE LOST

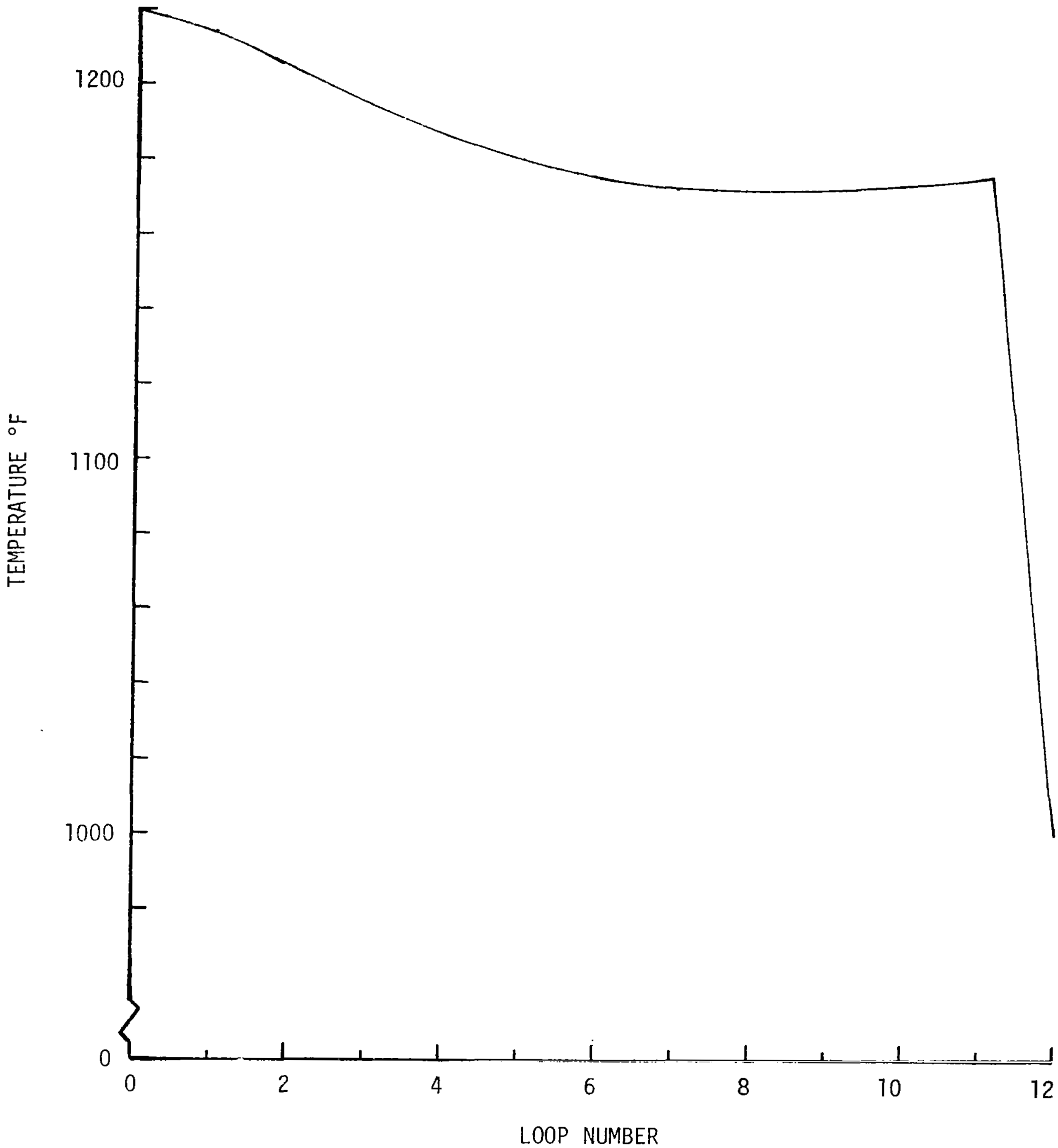


FIGURE 5-20 TEMPERATURE PROFILE WITH PUMP ONE LOST AND REDUCED HEAT REJECTION REQUIREMENT

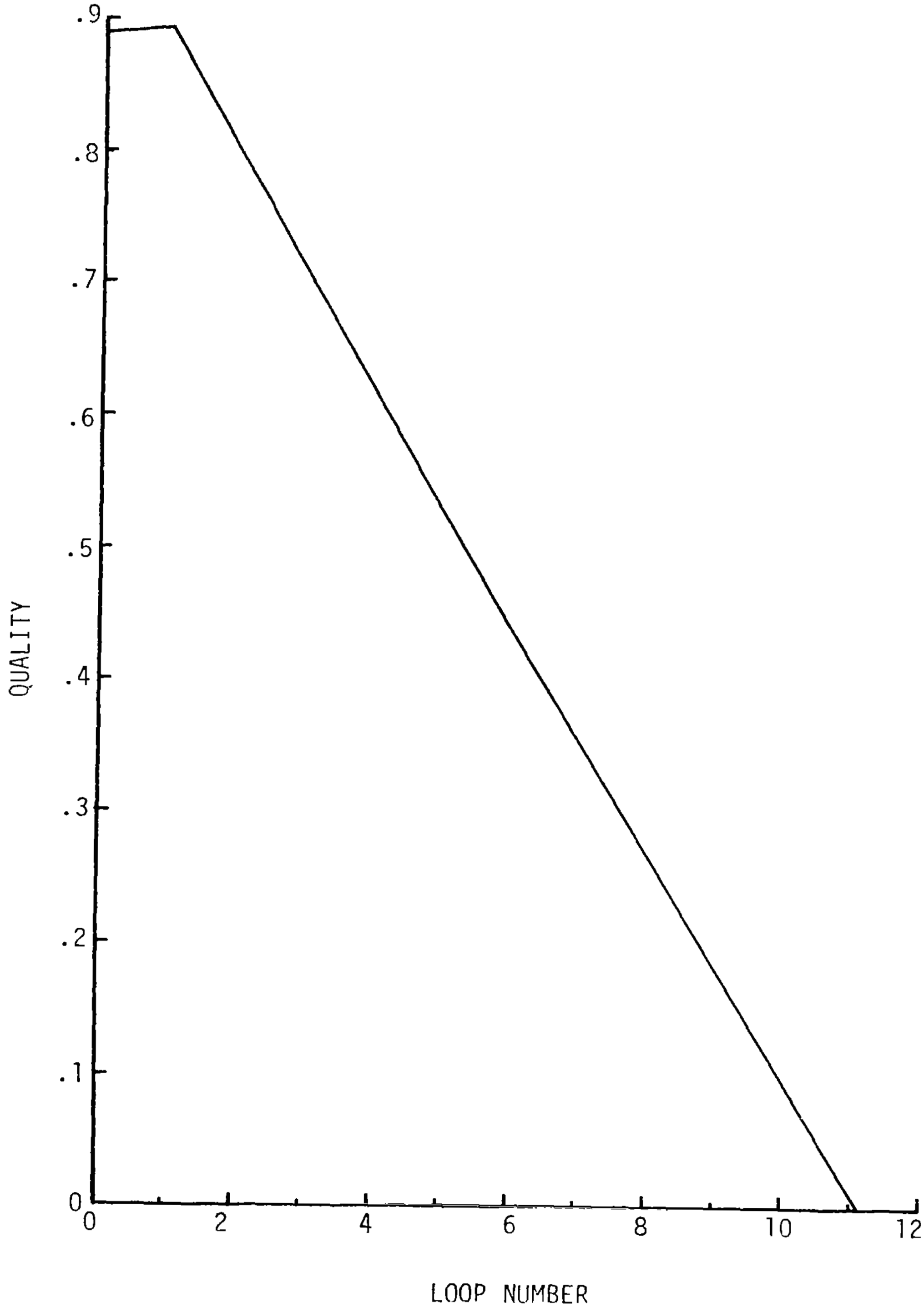


FIGURE 5-21 QUALITY PROFILE WITH PUMP ONE LOST AND REDUCED HEAT REJECTION REQUIREMENT

The principal reason for this effort was to determine the control scheme necessary to keep the inlet to the pump from being in the saturation region. The conclusions are:

- Twelve loops require immediate response and, therefore, automatic control system.
- Twenty-four loops or more can rely on operator or remote control
- At the point where a loss of two loops is acceptable, failure of one loop can be ignored and the operator can make corrections after two loops have failed.

6.0 SYSTEM OPERATIONAL PROBLEMS

A number of problem areas have been identified which may exist in the operation of a capillary pumped heat rejection system. Throughout the design process, an attempt was made to identify a design which was not only light in weight and practical from a fabrication standpoint but also tended to minimize these operational problems.

This section presents a discussion of potential problem areas indenti-fied. In general, rigorous solution of these problems is beyond the scope of the present study. Experimental work may be required to verify the potential solutions discussed here.

6.1 CONDENSING RADIATOR

Necessary to successful condensing radiator design is the ability to maintain a stable fluid flow subject to external or internal perturba-tions. Four flow instabilities pertinent to condensing radiators have been identified (Reference 13). These are interfacial instability, run-back instability, liquid leg instability and zero-g instability.

(a) Interfacial Instability

Interfacial instability is the inability of the fluid to form a stable meniscus which separates the condensing and subcooling portions of the radiator tube. The stability of the meniscus is dependent on surface tension energy and the kinetic and potential energy of a distur-bing wave. Interfacial stability can be maintained in the presence of external disturbances by selecting appropriate tube diameters. Using the criteria given in Reference 13, the radiator design of this study may experience interfacial instability for adverse accelerations exceeding 0.04 g. This problem is inherent with capillary pumped systems because of the small available pumping head and the resultant large radiator tube. Stabilizing this interface could possibly be done with appropri-ately placed tube inserts. The design and placement of these provisions would require extensive experimental study and is outside the scope of this program.

(b) Runback Instability

This type of instability is caused by low drag force exerted by the vapor flowing over the condensate. With an external body force, the vapor drag must be sufficient to accelerate the condensate to the outlet of the condenser. In zero-g operation, the condensate must be moved along the wall at a rate fast enough to prevent bridging of the tube. Using the criteria of Reference 13 applied to the last radiator segment, the present design could withstand an adverse acceleration of about 0.1 g. A possibility for increasing the vapor drag force is to taper the tube to keep the vapor velocity up as liquid is condensed.

(c) Liquid Leg Instability

This is caused by having a larger pressure gradient in the liquid film than in the vapor. External forces such as gravity can cause the static pressure in the liquid to be higher than the pressure in the vapor. The present design would withstand an adverse acceleration of approximately 0.04 g's before this condition would exist.

(d) Zero-G Instability

Zero-g instability is the emptying of liquid condensate into the vapor header due to a positive pressure gradient down the condensing tube. This condition is possible for short condensing length to diameter ratios when the two-phase frictional pressure drop is less than the momentum gain due to decelerating of the vapor flow when condensing. In this case, the interface static pressure would become larger than the inlet total pressure. The radiator design of this study has a static pressure drop of 0.12 psi (827 N/m^2) in the condensing tube section and this instability cannot exist.

The low pressure rise capabilities of a capillary pumper tend to make design of a stable radiator for any substantial gravity field difficult. Smaller tubes can be used in the radiator but only at a considerable weight penalty. Gravity forces along the tube length must be kept very low. This fact is likely to constrain the vehicle orientation. Any spacecraft which spins to provide artificial gravity must be

oriented such that the gravity force is perpendicular to the radiator flow path.

6.2 START-UP/SHUTDOWN

Start-up and shutdown of liquid metal heat rejection loops which are solid at normal environmental temperatures require careful consideration for any pumping concept; in the case of the passive capillary pumped system, the problem is particularly acute. This section describes some of the difficulties encountered in trying to stop and start a capillary pumped heat rejection system designed for a space environment and suggests one concept for alleviating these difficulties. It should be noted that the performance of capillary pumped systems, particularly for non-nominal conditions, is not well understood and the resolution of the start-up/shutdown operational problem must ultimately come through a careful experimental study where many of the unknowns and uncertainties can be simulated.

Start this discussion assuming that the power system is operating at full capacity and with inlet condensing potassium at the nominal 1220°F (933°K) and 0.89 quality. As the shutdown procedure begins, assume that this flow rate and temperature drops but that the quality remains approximately constant. The overall temperature level of the heat rejection system will drop with a corresponding decrease in the heat rejection capability. It has been shown in Section 5 that a substantial drop in the potassium inlet temperature results in a decrease in the operating pressure in the last capillary pumps until there is no longer sufficient pressure level to support the loop ΔP. Those pumps will fail to operate at or below this limit. This event occurs at fairly high temperature (~900°F, 755°K) so there is not immediate danger of freezing of the working fluid (~150°F, 339°K for potassium) when the loop fails.

Successful shutdown of the failed loop then requires a return of all the liquid to some convenient storage area. As a solution to this problem, it is proposed to include a coarse wick structure on the down-stream (vapor) side of the 400 mesh screen which provides the main capillary pressure rise. (See Figure 6-1.) During normal operation, the coarse

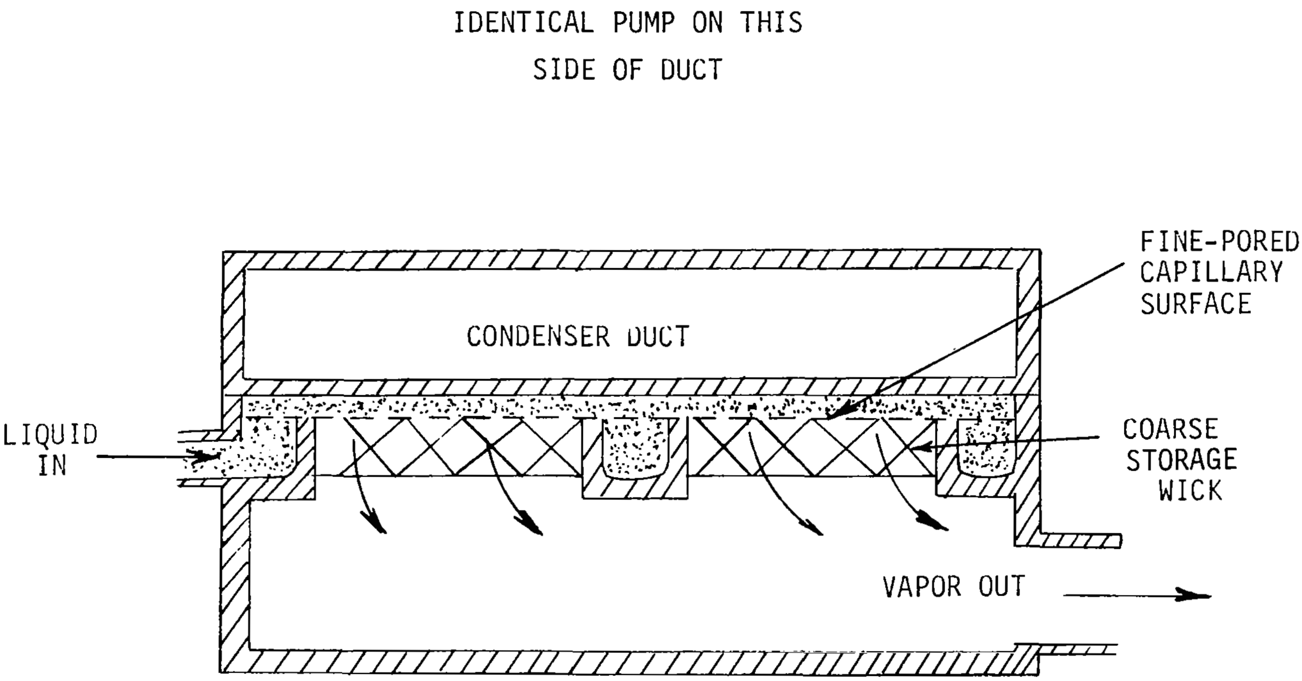


FIGURE 6-1. COARSE WICK STORAGE SCHEME

wick would be unable to sustain the capillary pressure head required and the liquid would recede back to the fine mesh and the course mesh would introduce little pressure drop in the system. In the static or isothermal condition, the coarse wick, having a pore size smaller than anything else in the loop (except the 400 mesh screen), will fill by emptying the liquid from the loop. How fast this will occur is very difficult to estimate. However, if the loop fails at 900°F (755°K), there will be a substantial time lag before there is dange of freezing.

The liquid inventory for a twelve loop configuration is approximately 0.3 ft^3 (.0085m^3). With a porosity of 70%, the required thickness of the coarse wick would be approximately 3/4 in. (.019m). However, it must be assumed at start-up that the radiator can be as low as -100°F (200°K) and that return of liquid back to pump will not occur until this temperature is increased to approximately 200°F (366°K). For a twelve loop case, this required 15000 Btu (15822000 Joules) of thermal energy to be supplied by condensing potassium vapor from the pump at 900°F (756°K). The coarse wick thickness must then be increased to 1.7 in. (.0432m) to provide for additional potassium storage required to heat the radiator up to a level which prevents freezing of the working fluid. No consideration was given to the process of introducing cold potassium into cold radiator tubes and the manner in which condensation/freezing occurs. This effect will obviously have a large influence on the applicability of this concept as well as on the design of the tubes.

A major concern with the scheme proposed,or in fact any scheme, is that there must be an excess liquid inventory to start the system from a frozen state and that provisions to remove this liquid must be made for normal operating conditions. The gas controlled liquid reservoir (see Figure 6-2) is one procedure for storage of excess liquid during normal operation. At shutdown, the coarse wick would empty liquid from the reservoir as well as the rest of the loop. A stop is provided on the bellows to prevent complete collapse at the low pressure frozen state. However, in normal operation, the accumulator would act to maintain constant system pressure and the reservoir would fill as the heat rejection loop approaches normal operating conditions.

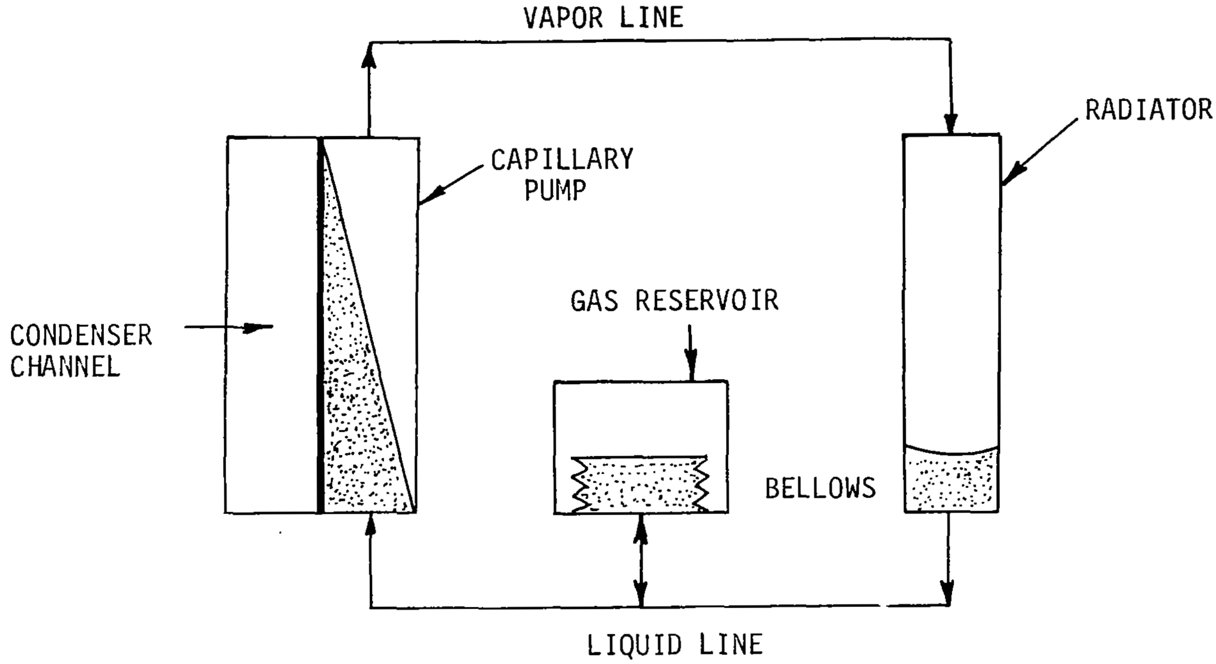


FIGURE 6-2. BELLOWS STORAGE

Another important characteristic of the coarse wick storage concept is that even in the frozen state, working fluid must be maintained in the gap between the condenser wall and the 400 mesh screen. Otherwise, there is no way to input heat into the coarse wick at start-up. Also, since the potassium superheat is limited before nucleate boiling occurs (approximately 150°F (83°K) was assumed in this study) the heat input to the coarse wick system must be limited until the liquid interface recedes back to the 400 mesh screen. Potassium of the thicknesses noted can tolerate approximately 6000 watts/ft^2 (64,500 watts/m^2) at 150°F (83°K) temperature difference.

Clearly the scheme presented to provide start-up and shutdown capability for capillary pumped heat rejection systems has many unknowns and uncertainties, most of which are very difficult to treat analytically.

6.3 NON-CONDENSIBLE GAS

The presence of non-condensible gases in the capillary pumped loop has the potential effect of causing the capillary surface to lose its pumping capacity. Preliminary experiments at TRW on capillary pumpers of the proposed configuration have shown that small amounts of gas at the screen surface can be tolerated but formation of large bubbles will eventually cause a collapse of the capillary head. It is relatively simple to prevent a limited amount of vapor/gas from reaching the pump. One such method is shown in Figure 6-3. A screen trap in the liquid line prior to the pump will trap the gas provided the working fluid wets the screen. Making the trap much larger in cross-section prevents an increase in the liquid side pressure drop which in turn would have substantial impact on the overall systems' optimization.

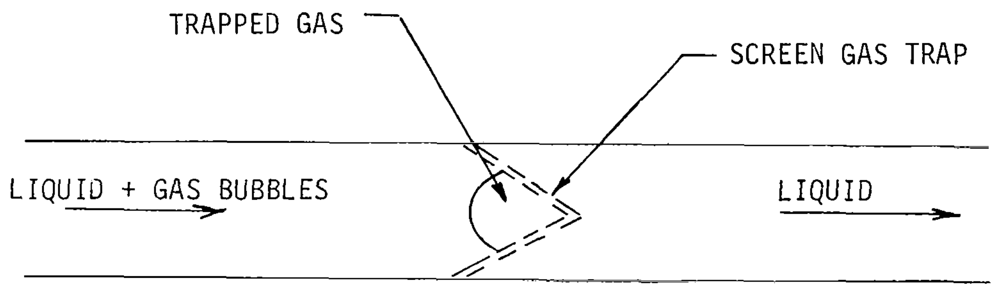


FIGURE 6-3. SCHEMATIC OF NON-CONDENSABLE GAS BUBBLE TRAP

A preliminary calculation was made for a four-loop heat rejection system to determine the magnitude of condensible gases introduced in the filling operation. Assuming evacuation to 10^{-6} mm Hg and 1 part in 10^7 trapped argon in the potassium charge, a gas volume of approximately 12 cc per loop can be expected at normal operating conditions. This volume represents, for a cone angle of 90°, a base diameter of approximately 1.8 in. (4.5 cm). The required overall diameter of the gas trap would be on the order of 2.5 - 3.0 in. to prevent large liquid side pressure drops. Note that this calculation is for a four loop system; the required volume for the 12 or the 24 loop configuration would be correspondingly smaller.

It is clear that continuous gas generation or diffusion of gas into the system cannot be dealt with by the trap shown in Figure 6-3. Should this condition represent a real situation, some positive means of gas venting is required (e.g., a non-wetting gas plug, a mechanical valve actuated by command or by a gas sensor). It seems apparent, however, that the only practical approach is to take action to prevent continuous gas buildup and to use the gas trap to deal with small amounts which cannot be avoided.

6.4 OPERATION IN GRAVITY

The capillary pumped heat rejection system described in Section 5 will not operate as configured in a one-g environment. To enforce the requirement for one-g operation would severely constrain the power system geometry in the case of a capillary pumped system. The liquid head of the proposed design is approximately 5.4 psi (37232 N/m^2) which is substantially beyond the maximum pumping capability of a 400 mesh screen wick. In fact, the height of the pump itself represents a pressure head of about 0.7 psi (4826 N/m^2). Ground operation would require that the pump/condenser assembly be lowered to the level of the liquid level in the radiator and, further, that the pump be oriented with the heat transfer area horizontal.

It should also be noted that ground operation of the radiator tubes as shown in Figure 5-10 provides a gravity drain of the condensate to the bottom of the tube. This condition would provide unrealistic performance data for the radiator. For meaningful ground testing of the heat rejection, it will be necessary to orient the radiator surface as well as

the pump in a horizontal position with the pump slightly elevated relative to the radiator.

It must be concluded that the capillary pumped heat rejection system imposes constraints on the ground testing of the unit. While other operational problems with this system may have total solutions, the ground testing difficulties can only be minimized by careful orientation of the capillary pumped loops relative to gravity and by a substantial amount of separate subsystem and element testing.

In a one-g environment the liquid would not bridge a 1.1 inch tube; thus the liquid condensate will flow along the bottom of the tube. It is possible some of the radiator stability problems could be alleviated by using smaller tube diameters.

7.0 CONCLUSIONS

Detailed conclusions pertaining to the specific design and to the performance trade-offs are presented in the appropriate sections. General conclusions which can be drawn from this study are the following:

- The capillary pumped concept is a feasible approach for the heat rejection loops of the Advanced Rankine Cycle Power System. This approach appears to offer some weight advantages over the conventional EM-pumped system.
- There are a number of system operational problems which constrain the application of capillary pumped units. These problems include start-up, shutdown, accumulation of non-condensible gases and sensitivity to gravity and other acceleration forces. However, these problems are not amenable to analytical treatment.
- The reliability of the capillary pumped system, while potentially greater than the EM-pumped unit, cannot be evaluated at this time. Reliability depends on a resolution to systems operating problems.
- The arterial fed screen wick capillary pump concept represents an attractive design approach for liquid metal systems.

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APPENDIX A

CONCEPTUAL DESIGN ANALYSIS

This appendix presents the analytical details used in evaluating various capillary pump wick configurations. The results of these analyses are summarized as Section 3.0 of the report, together with an evaluation which leads to the choice of a single screen wick displaced from the heat source by a uniform spacing for application to the ARCPS heat rejection system.

Section 1, Analysis of Stenger Type Cylindrical Capillary Pump, presents an analysis of the Stenger type wick configuration in which the heat is input to one side of a wick and liquid from the opposite side and the vapor is generated and carried away through cutouts in the wick between the hot wall and the wick material. Section 2, Analysis of Two-Dimensional Capillary Pump with Conventional Wick, gives an analysis of a flat conventional wick configuration in which heat is input on one surface, vapor generated on the opposite side and liquid is supplied through artery runners from the vapor side. Section 3, Analysis of a Two-Dimensional Screen Wick Capillary Pump, presents an analysis of a single screen wick separated from the hot surface by a uniform space configuration in which vapor is generated at the screen surface and liquid supplied by artery runners on the vapor side of the wick. Section 4, Analysis of a Three-Dimensional Screen Wick Capillary Pump, gives an analysis of the screen configuration just described with liquid feed by arteries running both horizontal and vertical directions to form a square liquid feed matrix.

1. ANALYSIS OF STENGER TYPE CYLINDRICAL CAPILLARY PUMP

1.1 INTRODUCTION

This section presents an analysis of a cylindrical capillary pump of the type used in the experimental study of Stenger (Reference 1). A schematic of this pump is presented in Figure A-1.

1.2 HEAT TRANSFER AREA

The conduction path for this pump is primarily through fins of length $(r_v - r_2)$. Let

α = (minimum fin cross-sectional area)/$2\pi r_2 L$

β = (conduction path length)/$(r_v - r_2)$

Then
$$Q = \frac{\alpha k \Delta T}{\beta (r_v - r_2)} (2\ r_2 L) = \dot{m}_\ell (r_2) h_{fg} \tag{A.1}$$

The symbols are defined in the nomenclature list. The mass flow rate can be written

$$\dot{m}_\ell = P_\ell \overline{V}(r) \int_0^{2\pi} L r d\theta = P_\ell \overline{V}(r)(2\pi r L) \tag{A.2}$$

and since $\dot{m}_r(r) = \dot{m}_r(r_2)$, equations (A.1) and (A.2) can be combined to give

$$\overline{V}\ell(r) = \frac{\alpha k \Delta T}{\beta P (r_v - r_2) h_{fg}} \left(\frac{r_2}{r}\right) \tag{A.3}$$

Darcy's law for this case is

$$dP/dr = \mu_\ell \overline{V}_\ell / K_p$$

Substituting equation (A.3) and integrating from r_1 to r_2 gives the pressure loss across the wick

$$(\Delta P)w = \frac{\alpha \mu_\ell k\ T r_2 \log(r_2/r_1)}{\beta K_p P (r_v - r_2) h_{fg}} \tag{A.4}$$

This result can be combined with equation (A.2) to give

$$\dot{m}_\ell = \frac{2\pi L P K_p (\Delta P)_w}{\mu_\ell \log(r_2/r_1)} \tag{A.5}$$

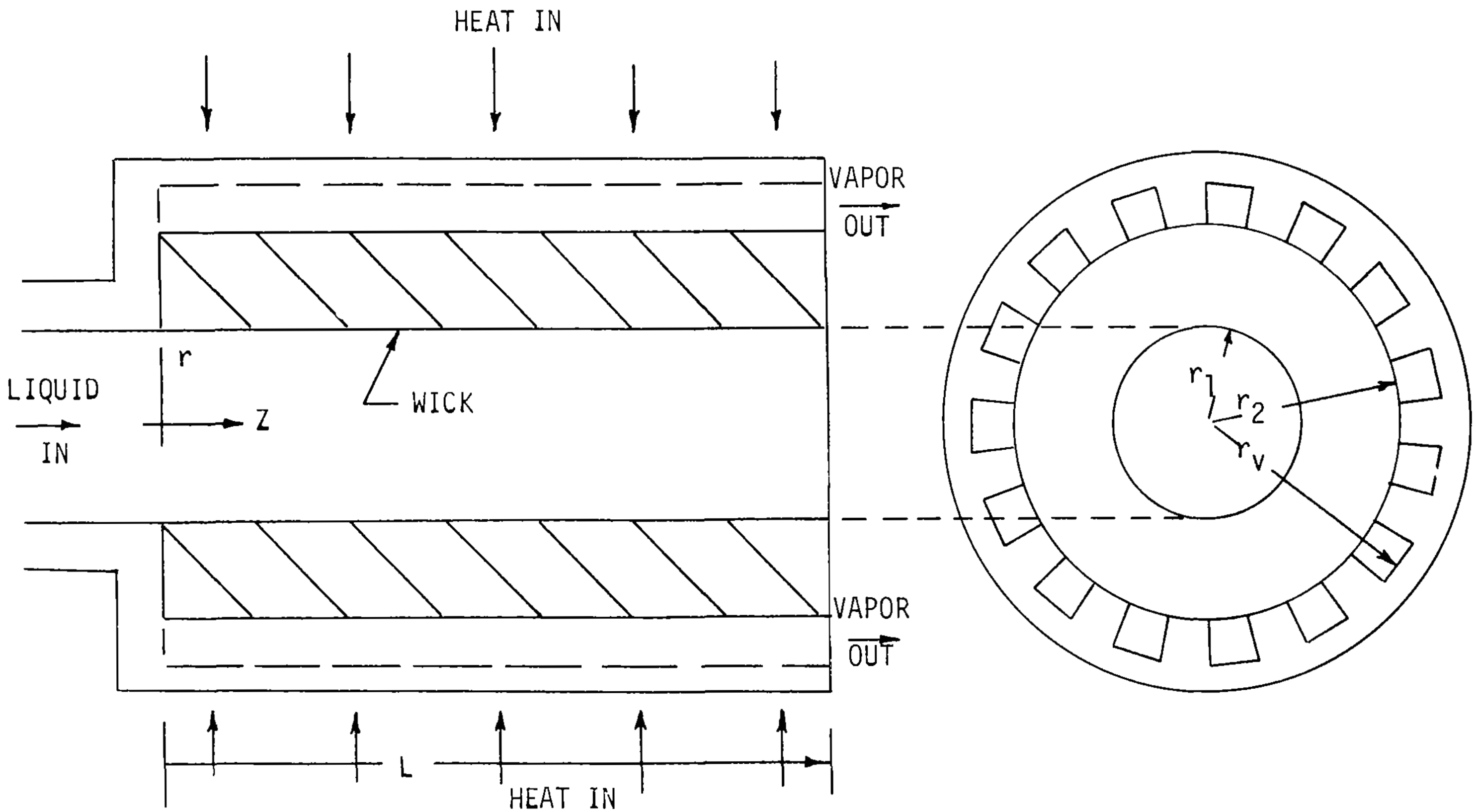


FIGURE A-1. SCHEMATIC OF CYLINDRICAL CAPILLARY PUMP

From equation (A.1), the heat flow rate is inversely proportional to the vapor space dimension $(r_v - r_2)$. The minimum allowable value of $(r_v - r_2)$ depends on the maximum allowable vapor flow velocity, and it is found (see Section 1.3) that this velocity is limited by the pressure head available to drive the vapor out of the pump. For the moment, it is assumed that the maximum allowable vapor flow velocity, $\overline{U}_{max}$, is known. Then if the fin surfaces lie on radii of the pump cross section,

$$\dot{m}_v = PV\overline{U}_{max}(1-\alpha)\pi(r_v^2 - r_2^2) \tag{A.6}$$

Since $\dot{m}_v = \dot{m}_\ell$, equations (A.5) and (A.6) can be combined to give

$$r_v = r_2 + \frac{2LPK_p(\Delta P)_w R_v T_v}{\mu_\ell P_v\overline{U}_{max}(1-\alpha)\log(r_2/r_1)} \tag{A.7}$$

The vapor density has been replaced by P_V/R_VT_V; i.e., the vapor is assumed to be a perfect gas. Now equations (A.4) and (A.7) can be combined to give

$$r_v^2 - r_2^2 = \left\{\frac{2\alpha kR_VT_V}{(1-\alpha)\beta P_v\overline{U}_{max}\, h_{fg}}\right\} \frac{r_2\, L\Delta T}{(r_v - r_2)} \tag{A.8}$$

This equation is cubic in r_v. However, $(r_v - r_2)$ will be substantially smaller than r_2 in most practical cases. Then $r_2 + r_v \approx 2r_2$ and equation (A.8) simplifies to

$$r_v - r_2 = \left\{\frac{\alpha kR_VT_VL\Delta T}{(1-\ )\, P_v\overline{U}_{max}\, h_{fg}}\right\}^{1/2} \tag{A.9}$$

Finally, the total heat transfer area for N pumps is $2\pi r_2 LN$, and from equations (A.1) and (A.9)

$$A = Q\left\{\frac{\beta R_VT_VL}{\alpha(1-\alpha)P_v\overline{U}_{max}h_{fg}\ k\Delta T}\right\}^{1/2} \tag{A.10}$$

As expected, the area is minimized by maximizing the fin thermal conductivity and the vapor flow velocity. The pump is like the thin-screen pump (Section 3) in that $A \sim L^{1/2}$, but in this case the $L^{1/2}$ dependence arises from friction in the vapor flow rather than friction in the liquid flow. Note also that A does not depend on the wick properties provided that the wick conductivity and the conducitivity of the working fluid are not much different than the fin conductivity. Of course the wick must be able to

supply the required mass flow rate. For the designs considered here (see Section 1.5) the pressure loss in the wick is a small fraction of the total pump pressure loss.

1.3 MAXIMUM VAPOR VELOCITY

The total pump pressure head (capillary pressure) is

$$2\sigma\cos\theta/r_c$$

For the smallest value of r_c ($.12 \times 10^{-3}$ ft.) for the wick materials in Table 10 of Reference 3, the total capillary head in Potassium at 1200°F is 84 psf. The radiator loss is assumed to be 72 psf, hence the pump loss must be less than 12 psf.

For the designs considered here, the total pressure loss in the liquid flow within the pump is negligible (see Section 1.5). Consequently, the available head can be used to size the vapor flow space.

The vapor space consists of a rectangular channel, one of whose walls is injecting fluid into the flow. This flow is rather complex, however, the results given in Reference 3 (for laminar, incompressible pipe flow with injection* through a porous wall) are probably adequate for this analysis. The pressure gradient is given by

$$\frac{dP_v}{dz} = -\frac{8\ \nu\bar{U}_v}{r^2}\left(1+3/4\ R_{e_r} - \frac{11}{270} R_{e_r}^2 + \ldots\right) \qquad R_{e_r} \ll 1 \qquad \text{(A.11)}$$

$$\frac{dP_v}{dz} = -\frac{\pi^2}{4} P_v\bar{U}_v \frac{d\bar{U}_v}{dz} \qquad R_{e_r} \rightarrow \infty \qquad \text{(A.12)}$$

where:
$$R_{e_r} = P_v r\ \bar{V}_v/\mu_v \qquad \text{(A.13)}$$

The result for small R_{e_r} is the same as for Poiseuille flow except for the power series in R_{e_r}. For large R_{e_r}, the pressure drop results primarily from the acceleration of the flow.

*Injection from <u>all</u> portions of the wall surface.

The vapor space is expected to be approximately square, hence $(r_v - r_2)$ is approximately equal to the hydraulic diameter of the channel. From continuity, $\rho_v \overline{V}_v = \rho_\ell \overline{V}_\ell$. Then, equations (A.3) and A.13) can be combined to give

$$R_{e_r} = \alpha k \Delta T / \beta \mu_v h_{fg} \tag{A.14}$$

For $\alpha = .5$, $\beta = 1.5$, and Potassium at 1200°F as the working fluid, the smallest and largest values of R_{e_r} of interest here are, respectively

$$R_{e_r} = .21 \quad \Delta T = 5°F,\ k = 10 \ \frac{BTU}{hr\ ft\ °R} \ \text{(stainless steel)}$$

$$R_{e_r} = 15 \quad \Delta T = 50°F,\ k = 71 \ \frac{BTU}{hr\ ft\ °R} \ \text{(tungsten)}$$

The quantity R_{e_r} is not clearly much greater than or much less than unity. In any case, and lacking better information, the equation giving the largest pressure gradient is used here. Moreover, the correction to the Poiseuille pressure gradient will be neglected, because this makes the analysis much simpler and because it represents a small correction in the range where it is presumably valid. Note that this procedure is unconservative; i.e., one of the two contributions (viscous or momentum) to the pressure drop is neglected.

If the evaporation is uniform

$$\overline{U} = \overline{U}_{max} \ (Z/L)$$

and the pressure difference in the vapor (along a path from Z=0 to Z=L) is given by

$$(\Delta P)_v = \frac{16 \mu_v \overline{U}_{max} L}{(r_v - r_2)^2} \qquad Re_r \to 0$$

$$(\Delta P)_v = \frac{\pi^2 P_v \overline{U}_{max}}{8} \qquad Re_r \to \infty$$

The consequences of these results for Potassium at 1200°F and $(\Delta P)_v$=12 psf are

$$\overline{U}_{max} = 2 \times 10^6 (r_v - r_2)^2/L \quad \text{ft/sec} \qquad R_{e_r} \to 0$$

$$\overline{U}_{max} = 200 \text{ ft/sec} \qquad R_{e_r} \to \infty$$

These results are identical for $(r_v - r_2)^2/L = 10^{-4}$ ft.; e.g., $r_v - r_2 = 10^{-2}$ ft. (~1/8") and L = 1 foot. The Reynolds number of the vapor flow in the axial direction is

$$Re_v = 14.6$$

based on $\overline{U}_{max}$ = 200 ft./sec. and $(r_v - r_2) = 10^{-2}$ ft. Hence, laminar flow relations are appropriate. For $(r_v - r_2)^2/L < 10^{-4}$ ft., equation (A.8) becomes

$$r_v^2 - r_2^2 = \left\{ \frac{2\alpha k R_v T_v}{(1-\alpha)\beta P_v f_o h_{fg}} \right\} \frac{r_2 L^2 \Delta T}{(r_v - r_2)^3} \qquad \text{(A.8a)}$$

where: $f_o = 2 \times 10^6$ 1/sec

Again making the assumption

$$r_2 + r_v = 2r_2$$

equations (A.9) and (A.10) become

$$(r_v - r_2) = \left\{ \frac{\alpha k R_v T_v L^2 \Delta T}{(1-\alpha)\beta P_v f_o h_{fg}} \right\}^{1/4} \qquad \text{(A.9a)}$$

$$A = Q \left\{ \frac{\beta^3 R_v T_v L^2}{\alpha^3 (1-\alpha) P_v f_o \; f_g (k\Delta T)^3} \right\}^{1/4} \qquad \text{(A.10a)}$$

Vapor space thickness, $(r_v - r_2)$, is presented in Figure A-2 versus L and ΔT for Potassium at 1200°F as the working fluid, $\alpha = .5$, $\beta = 1.5$, and for two fin materials: Stainless Steel and Tungsten. Total heat transfer area for $Q = 5.24 \times 10^6$ BTU/hr. is presented in Figures A-3 and A-4 for Stainless Steel and Tungsten fins, respectively. Note that $(r_v - r_2)^2/L < 10^{-4}$ ft. for all the temperature differences plotted if the fins are Stainless Steel. Thus, equations (A.9a) and (A.10a) were used. For Tungsten fins, $(r_v - r_2)^2/L > 10^{-4}$ ft. for $\Delta T \gtrsim 11$°F, hence, equations (A.9) and (10), with $\overline{U}_{max}$ = 200 ft.sec., were used for the four higher temperature differences.

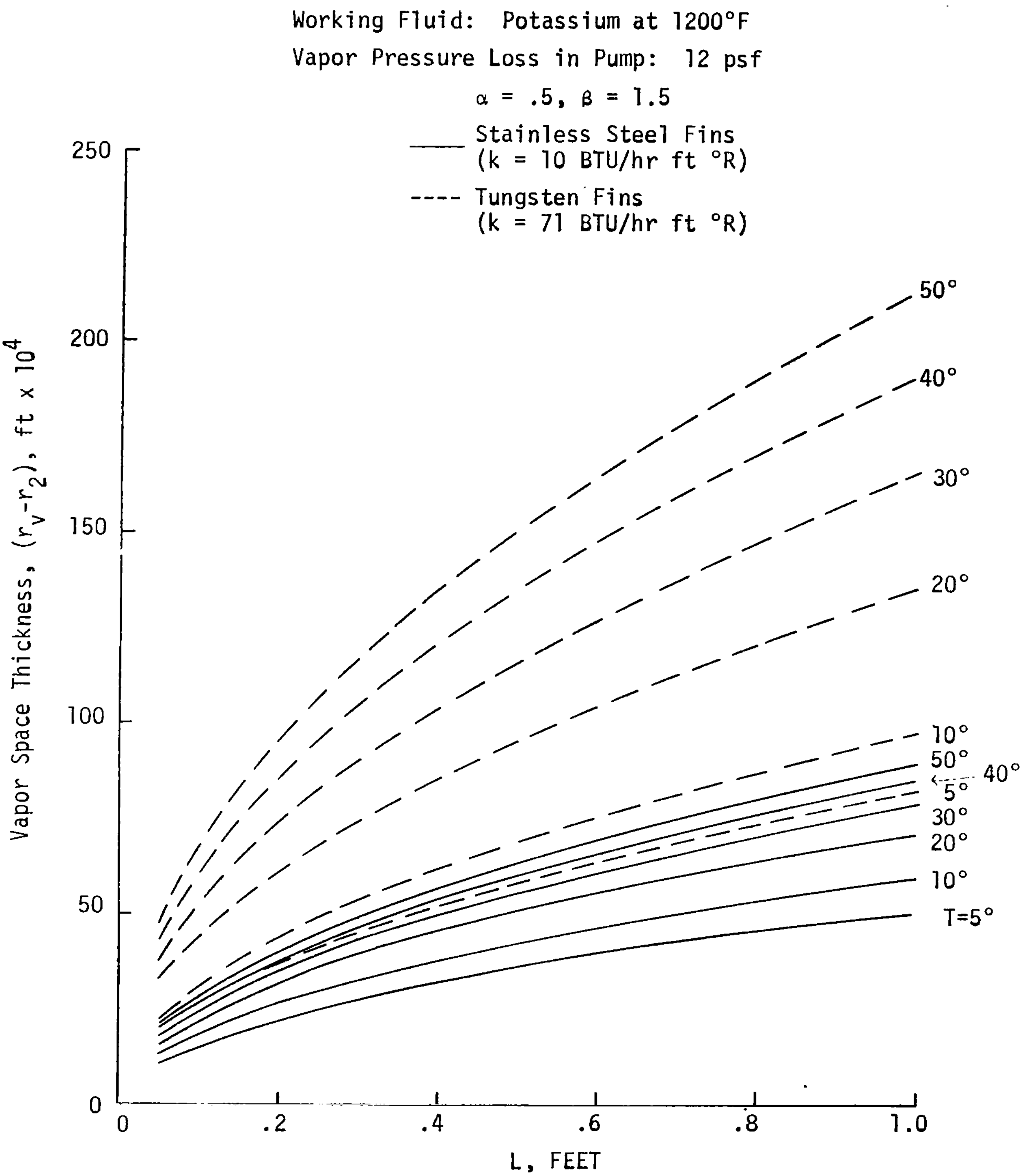


FIGURE A-2. CAPILLARY PUMP VAPOR SPACE THICKNESS VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES

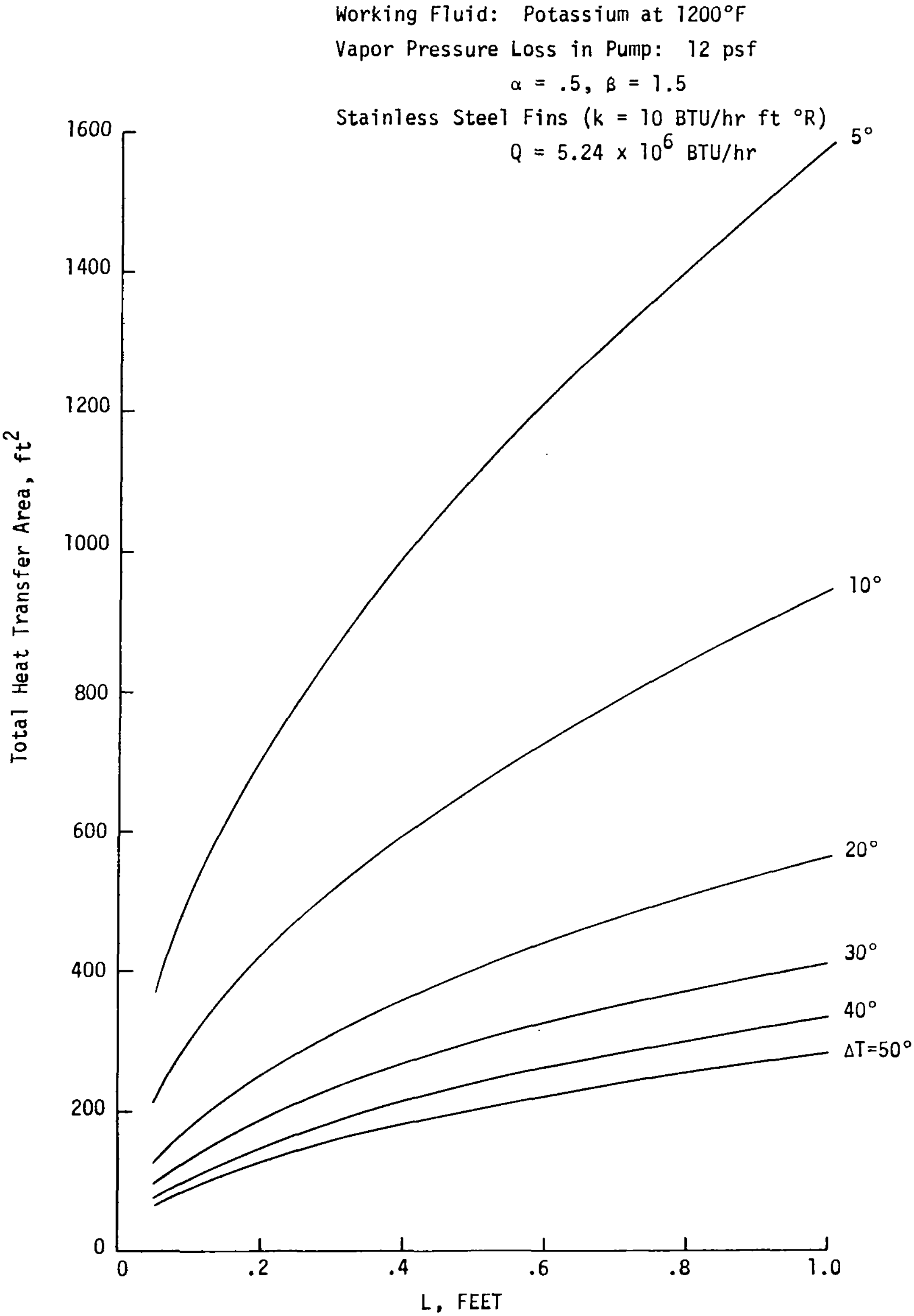


FIGURE A-3. CAPILLARY PUMP HEAT TRANSFER AREA VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES (STAINLESS STEEL FINS)

Working Fluid: Potassium at 1200°F

Vapor Pressure Loss in Pump: 12 psf

$\alpha = .5$, $\beta = 1.5$

Tungsten Fins (k = 71 BTU/hr ft °R)

$Q = 5.24 \times 10^6$ BTU/hr

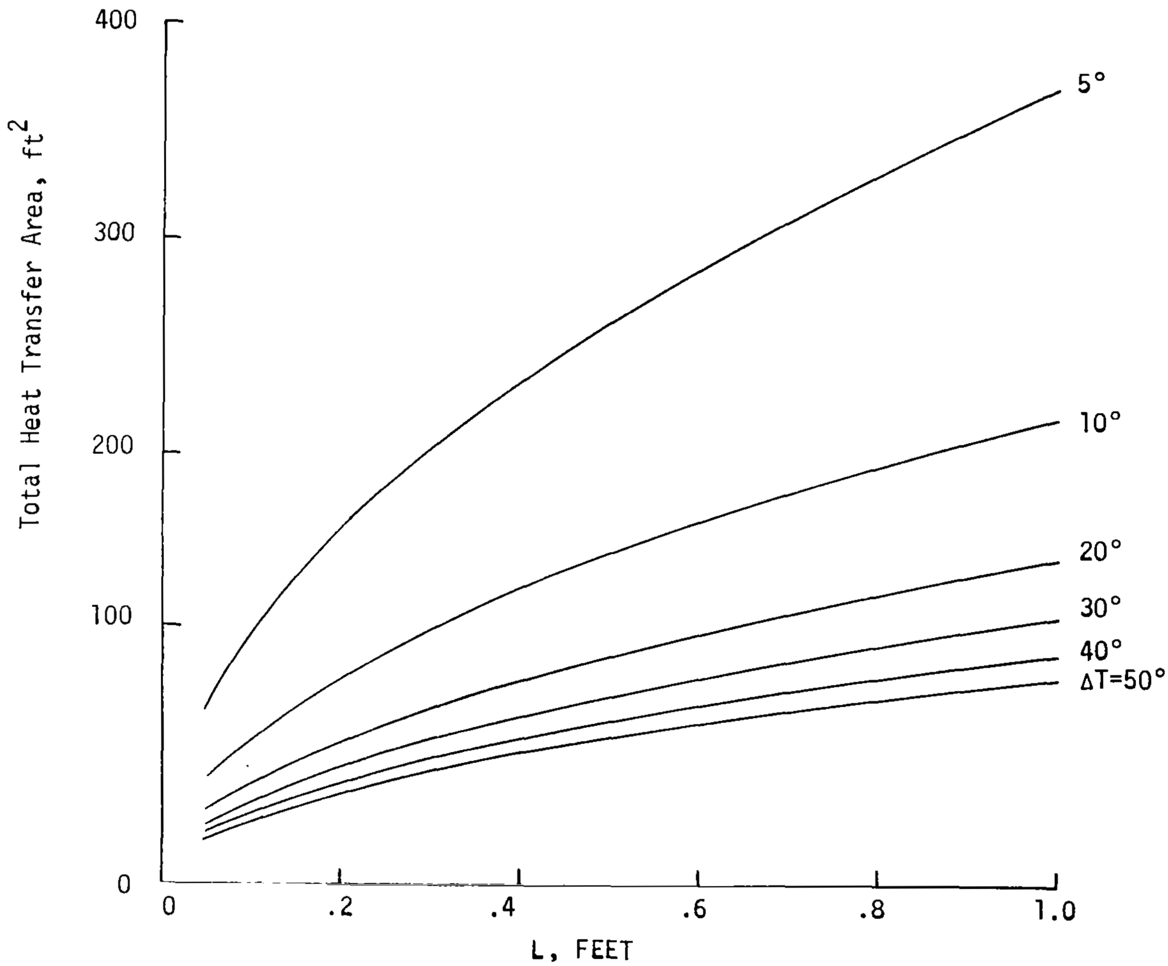


FIGURE A-4. CAPILLARY PUMP HEAT TRANSFER AREA VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES (TUNGSTEN FINS)

1.4 MAXIMUM INTERNAL DIAMETER OF WICK; NUMBER OF PUMPS

It is desirable that the liquid flow within the wick feed tube be in the form of a slug which fills the tube cross-section rather than a pool. This implies that the capillary pressure head in the liquid supply duct be greater than the difference in hydrostatic pressure across the duct.

Then $$P_\ell g d_1 < 4\sigma\cos\theta/d_1 \tag{A.15}$$

For operation on earth with Potassium at 1200°F as the working fluid, equation (A.15) implies that

$$d_1 < .0209 \text{ ft.}$$

This requirement results in a large number of pumps being necessary to obtain the required heat transfer area. The number of pumps required is

$$N = \frac{A}{2\pi r_2 L} \tag{A.16}$$

Consider $r_2 = d_1 = .02$ ft. $(r_2/r_1=2)$. Then for the <u>smallest</u> value of A/L in Figure A-4 (94.7 ft. at L = 1 ft., $\Delta T = 50°F$)

$$N = 755$$

One is led to consider an annular liquid space. However, if r_2 is large enough so that N is reasonable, the pump wick is locally flat, and the pump is like the thin screen pump except that the wick is thicker and the heat is supplied from the vapor side.

1.5 LIQUID FLOW PRESSURE LOSSES

In this subsection, it is confirmed that the pressure loss in the liquid flow within the pump is negligible. The total mass flow rate is

$$\dot{m}_T = \frac{Q}{h_{fg}} = 1.83 \ \frac{\text{lbm}}{\text{sec.}}$$

The largest liquid flow rate in an individual pump corresponds to the case with the smallest number of pumps (755 from Section 1.4), viz.,

$$2.42 \times 10^{-3} \text{ lbm/sec.}$$

The mean flow velocity at the entrance to the pump is .167 ft/sec (for d_1 = 0.02 ft) and the corresponding pressure gradient in Poiseuille flow is .0411 lbf/ft^3.

The largest pressure loss in the wick occurs for the smallest area rather than the smallest A/L. If equation (A.5) is multiplied by N, $\dot{m}_T$ replaced by Q/h_{fg}, and use made of equation (A.16), there is obtained

$$(\Delta P)_w = \frac{\mu_\ell r_2 \log(r_2/r_1) Q}{P_\ell K_p A h_{fg}}$$

For the smallest area from Figure A-4 (21.2 ft^2), r_2 = .02 ft., r_2/r_1 = 2, and K_p = .0472 x 10^{-8} ft^2 (for Hl Nickel felt from Table 10 of Reference 3)

$$(\Delta P)_w = .168 \text{ lbf/ft}^2$$

Finally, we can evaluate the error due to replacing $(r_v + r_2)$ in equation (A.8) by $2r_2$. The largest error occurs for ΔT = 50°F, L = 1 ft, and with Tungsten fins. If r_2 = .02 ft., the error in $(r_v - r_2)$ and A is about 20 percent. For the Stainless Steel fins, the largest error is about 5 percent. This error is conservative; i.e., the correct area is smaller. However, the neglect of one of the contributions to the vapor pressure gradient is unconservative, and the results in Figures A-3 and A-4 are expected to be somewhat unconservative.

2. ANALYSIS OF A TWO-DIMENSIONAL CAPILLARY PUMP WITH CONVENTIONAL WICK

The case of a thick wick which fills the space between heat source and vapor is considered in this section. Figure A-5 presents a schematic of a capillary pump cell of this type.

The fluid motion is described by the continuity equation

$$-\frac{d\overline{V}_z}{dz} = \frac{k\Delta T}{Ph^2\, h_{fg}} \tag{A.17}$$

and Darcy's law

$$\frac{dP}{dz} = -\frac{\mu}{K_p}\,\overline{V}_z \tag{A.18}$$

Integrating equation (A.17) subject to the boundary condition that $\overline{V}_z = 0$ at z = 0 and substituting in equation (A.18) yields

$$\frac{dP}{dz} = \frac{\mu k\Delta T z}{PK_p h^2\, h_{fg}} \tag{A.19}$$

Then the pressure difference between z = 0 and z = L is given by

$$\Delta P = \frac{\mu k\Delta \dot{T} L^2}{2PK_p h^2\, h_{fg}} \tag{A.20}$$

The total heat flow rate through an element of length L and width W is given by

$$Q = \frac{k\Delta TWL}{h} \tag{A.21}$$

From equations (A.20) and (A.21), the pump area required to transfer a given heat flow rate is

$$A = QL\left\{\frac{\mu}{2\rho\, k\Delta TK_p\Delta Ph_{fg}}\right\}^{1/2} \tag{A.22}$$

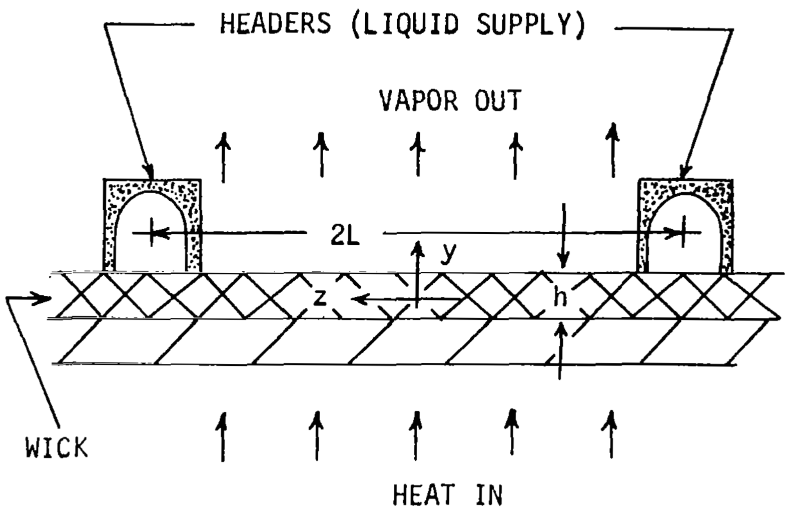


FIGURE A-5. SCHEMATIC OF CAPILLARY PUMP CELL

It is assumed for the purpose of this analysis that the pressure loss in the condenser is 1/2 psi (72 lbf/ft^2). The pressure head required to drive the vapor through the evaporator is negligible, then we also have

$$\Delta P = \frac{2\sigma \cos\Theta}{r_c} - 72\ lbf/ft^2 \qquad (A.23)$$

where: r_c = effective pore radius

Θ = wetting angle

σ = surface tension

It is obviously necessary that the capillary head be greater than 72 lbf/ft^2. For potassium at 1200°F ($\sigma = 50.4 \times 10^{-4}$ lbf/ft) wetting most metals ($\Theta \approx 0$), this implies that

$$r_c < .14 \times 10^{-3}\ ft. \qquad (A.24)$$

It is also desirable that the product $K_p \Delta P$ be as large as possible. This results in a minimum thickness, h, from equation (A.20), for given ΔT, L, and liquid properties, and a minimum area, from equation (A.22), required for a given heat flow rate. Of three materials which satisfy equation (A.24)

400 mesh screen
H1 and H3 nickel felt (Reference 2, Table 10)

the maximum value of $K_p \Delta P$ is

$$(12\ psf)(.0472 \times 10^{-8} ft^2) = .566 \times 10^{-8}\ lbf$$

for H1 nickel felt.

For this wick material and for Potassium at 1200°F as the working fluid, Figure A-6 presents h versus L and ΔT. Figure A-7 presents L and ΔT for the further specification that $Q = 5.24 \times 10^{-6}$ BTU/hr.

Working Fluid: Potassium at 1200°F
Wick Material: H1 Nickel Felt
Flow Pressure Loss in Wick: 12 psf
Two-Dimensional Analysis (One-Dimensional Flow)

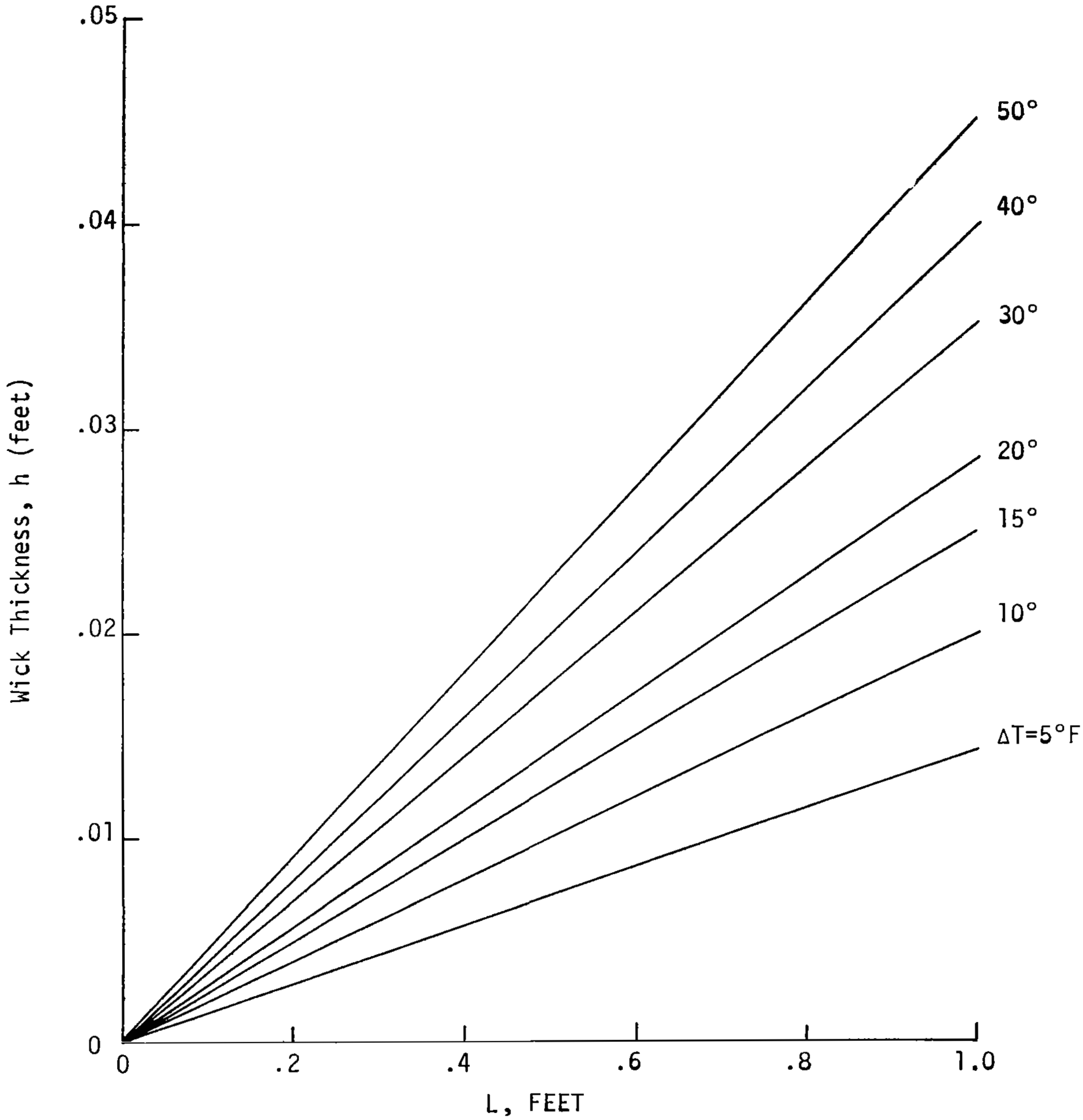


FIGURE A-6. CAPILLARY PUMP WICK THICKNESS VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES ACROSS WICK

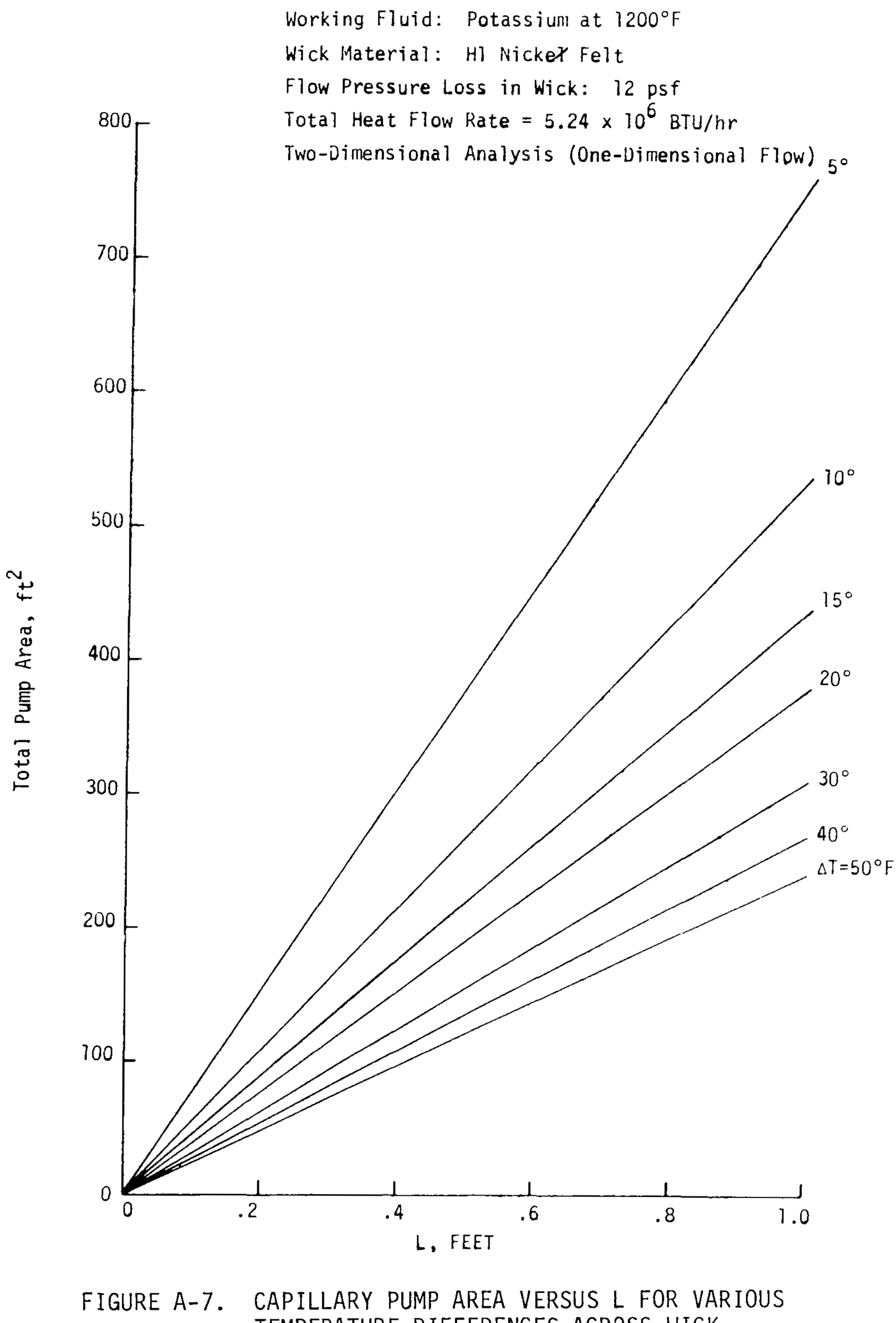
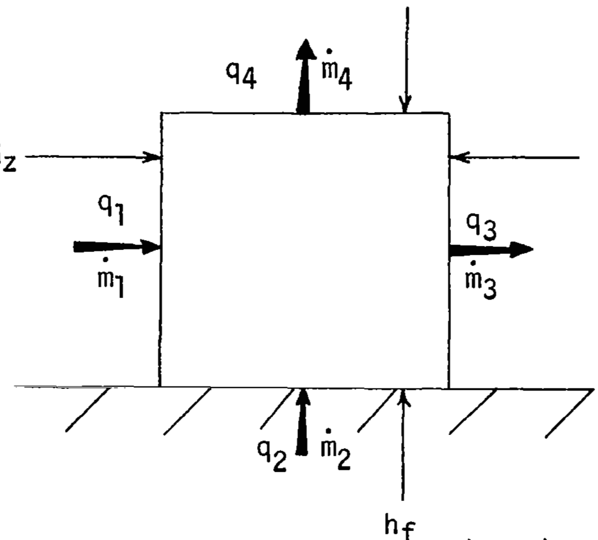


FIGURE A-7. CAPILLARY PUMP AREA VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES ACROSS WICK

3. ANALYSIS OF TWO-DIMENSIONAL CONSTANT-GAP CAPILLARY HEAT PUMP

The following section analyzes the use of a single screen wick which is held at a uniform distance from the heat source. The screen is fed by liquid flow through this gap from line sources on two opposite edges.

Consider an element of fluid of length dz, width w, and height h_f.



Assume: $P = \text{const}$

$q_1 = q_3$

$T_2 - T_4 = \Delta T$

Since $\dot{m}_2 = 0$

$$\dot{m}_4 = \dot{m}_1 - \dot{m}_3$$

$$= \rho W h_f[\bar{V}_z(Z) - \bar{V}_z(z+dz)]$$

or

$$\dot{m}_4 = \rho W h_f \frac{d\bar{V}_z}{dz}\, dz \qquad \text{(A.25)}$$

where: $\bar{V}$ is an average velocity defined by $\bar{V}$ = total mass flow/flow area. The conduction length includes the screen thickness, h_s. Thus,

$$q_2 = q_4 = k \frac{\Delta T}{(h_f+h_s)} = k \frac{\Delta T}{h_t}$$

The total rate of heat transport in the vertical direction is

$$k \frac{\Delta T}{h_t} Wdz = \dot{m}_4\, h_{fg} \qquad \text{(A.26)}$$

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Then from (A.25) and (A.26)

$$\frac{d\bar{V}_z}{dz} = -\frac{k\,\Delta T}{\rho h_f\, h_t\, h_{fg}} \qquad \text{(A.27)}$$

At $z=0$, $\overline{V}=0$ (see Figure A-8), then

$$\overline{V}_z = \frac{k\ \Delta T z}{\rho h_f\ h_t\ h_{fg}} \tag{A.28}$$

There is a small velocity component in the y-direction. Since

$$\dot{m} = \rho V A$$

we have from (A.26) that

$$\overline{V}_y = \frac{k\ \Delta T}{\rho h_f\ h_{fg}}$$

This is smaller than $\overline{V}_z$ by the ratio h_f/L.

The velocity is related to the pressure gradient by the z-momentum equation:

$$0 = \frac{\partial P}{\partial z} + \mu \frac{\partial^2 V_z}{\partial^2 y} \tag{A.29}$$

It is assumed that the convective terms

$$\rho(V_z \frac{\partial V_z}{\partial z} + V_y \frac{\partial V_z}{\partial y}$$

are negligible. This will be considered again. The boundary conditions for equation (A.29) are

$$V_z = 0 \text{ for } y = h_f/L$$

and the solution is:

$$V_z = \frac{1}{2\mu} \frac{\partial P}{\partial z} (y^2 - \frac{h_f^2}{4})$$

In terms of an average velocity,

$$\overline{V} = \text{total mass flow/flow area}$$

the solution is:

$$\frac{dP}{dz} = -\frac{12\mu\overline{V}}{h_f^{\ 2}} \tag{A.30}$$

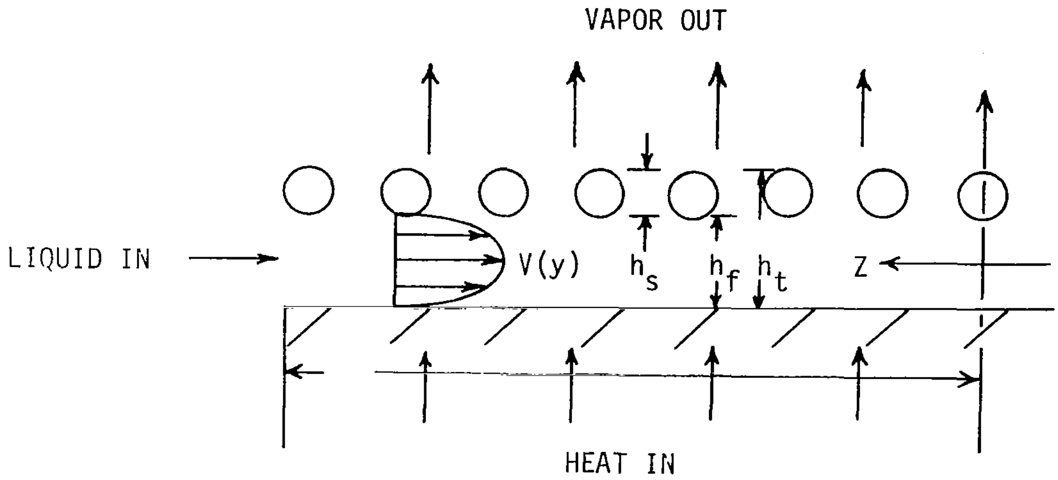

FIGURE A-8. SCHEMATIC OF CAPILLARY HEAT PUMP

Combining equations (A.28) and (A.30) yields

$$\frac{dp}{dz} = \frac{12\mu k\ \Delta T\ z}{\rho h_f\ h_f^2\ h_{fg}}$$

then $$\Delta P = \frac{12\mu k\ \Delta T}{\rho h_f\ h_f^3\ h_{fg}} \int_0^L z dz = \frac{6\mu k L^2\ \Delta T}{h_f\ h_f^3\ h_{fg}}$$

and $$h_t = \frac{6\mu k L^2\ \Delta T}{\rho h_f^3\ h_{fg}\ \Delta P} \tag{A.31}$$

For negligible screen thickness

$$h_t = h_f = h$$

and $$h = \left\{\frac{6\mu k \Delta T L^2}{\rho h_{fg}\ \Delta P}\right\}^{1/4} \tag{A.32}$$

Now $$dQ = qwdz = k\frac{\Delta T}{h_t}\ wdz$$

and $$Q = \frac{dQ}{dz}dz = \frac{k\Delta TWL}{h_t}$$

since $Q(0) = 0$. This can be rewritten

$$Q = \frac{k\Delta TWL}{h}\ \frac{h}{h_t} \tag{A.33}$$

From equations (A.31) and (A.32),

$$\frac{h}{h_t} = \left(\frac{h_f}{h}\right)^3$$

~~or~~ $$\cancel{h = h_f^3 (h_f + h_s)} \tag{A.34}$$

~~since $h_t = h_f + h_s$.~~ Then equation (A.33) becomes

$$Q = \frac{k\Delta TWL}{h}\left(\frac{h_f}{h}\right)^3 \tag{A.35}$$

Defining

$$Q_o = \lim_{h_f \to h} Q = \frac{k\Delta TWL}{h}$$

then

$$\frac{Q}{Q_o} = \left(\frac{h_f}{h}\right)^3 \qquad \text{(A.36)}$$

Alternatively, if Q is fixed and the required area (WL) is sought, we define

$$A_o = \lim_{h_f \to h} A = \frac{hQ}{k\Delta T}$$

and from equation (A.35)

$$\frac{A}{A_o} = \left(\frac{h}{h_f}\right)^3 \qquad \text{(A.37)}$$

Equations (A.34) and (A.37) can be used to calculate the effect of any given screen height, h_s, on the required area and flow thickness. Figure A-9 presents h_f and A/A_o versus h for h_s = 4 mils. Figures A-10 and A-11 present h_f and A = WL, respectively, versus L and ΔT for Potassium at 1200°F as the working fluid, ΔP = 81.2 lbf/ft^2, h_s = 4 mils, and Q = 5.24 x 10^6 Btu/hr.

This analysis is valid only if the convective terms in the z-momentum equation are negligible and if the flow is laminar. Both of these conditions fail if the velocity is high enough. From equation (A.28) the maximum velocity (at z=L) is given by

$$V_z(L) = \frac{k\Delta TL}{\rho h_f \, h_t \, h_{fg}}$$

For simplicity, consider $h_f = h_t = h$. Then, using equation (A.32),

$$V_z(L) = \left(\frac{k\Delta T\Delta P}{6\mu T \, h_{fg}}\right)^{1/2} \qquad \text{(A.38)}$$

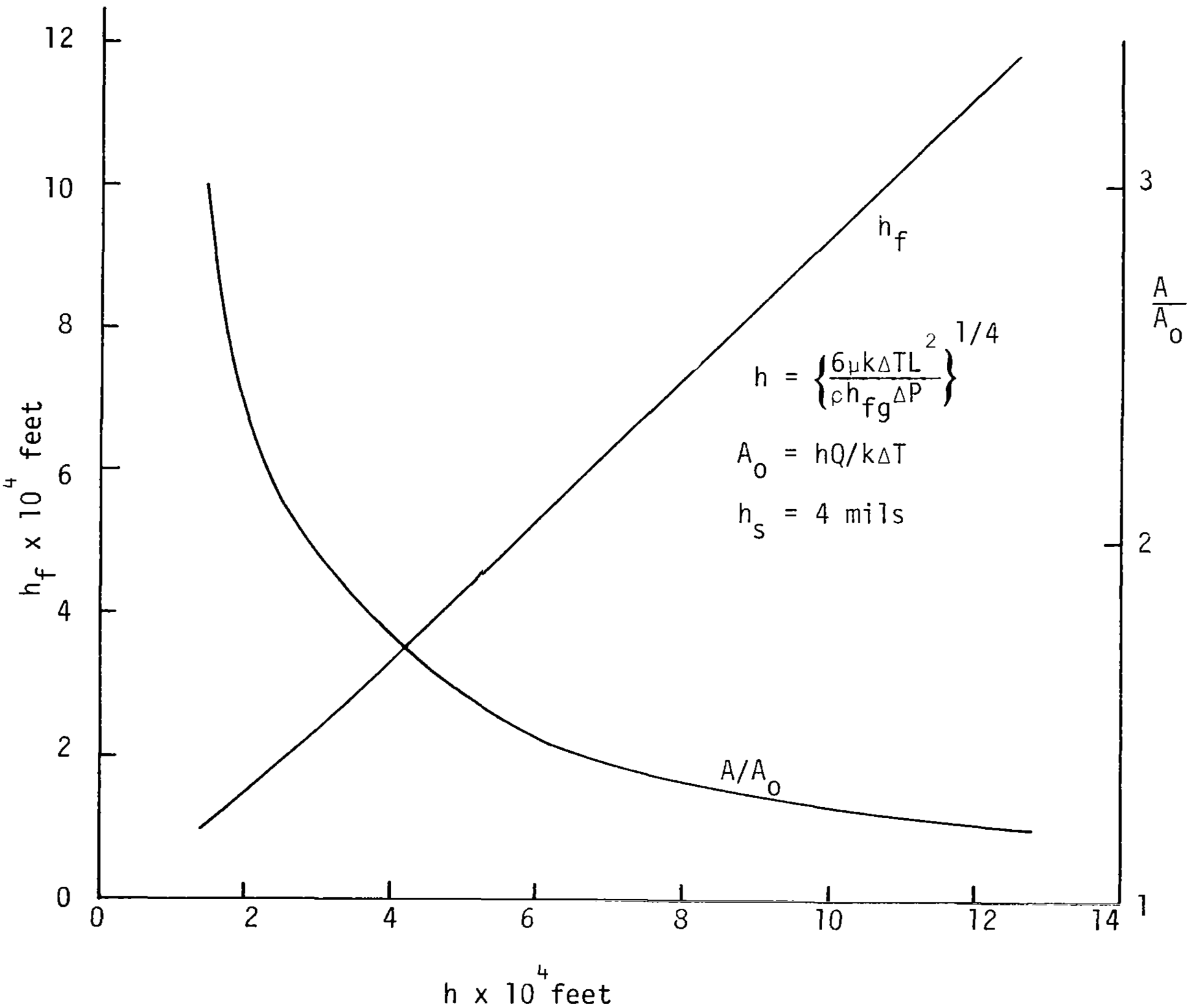

FIGURE A-9. CAPILLARY PUMP GAP THICKNESS, h_f, AND RATIO OF REQUIRED AREA FOR h_s = 4 mils TO REQUIRED AREA FOR h_s = 0, A/A_o, VERSUS h (Gap Thickness for h_s = 0)

WORKING FLUID: Potassium at 1200°F

Pump Pressure Head = 81.2 lbf/ft^2

Screen Thickness = 4 mils

Two-Dimensional Analysis

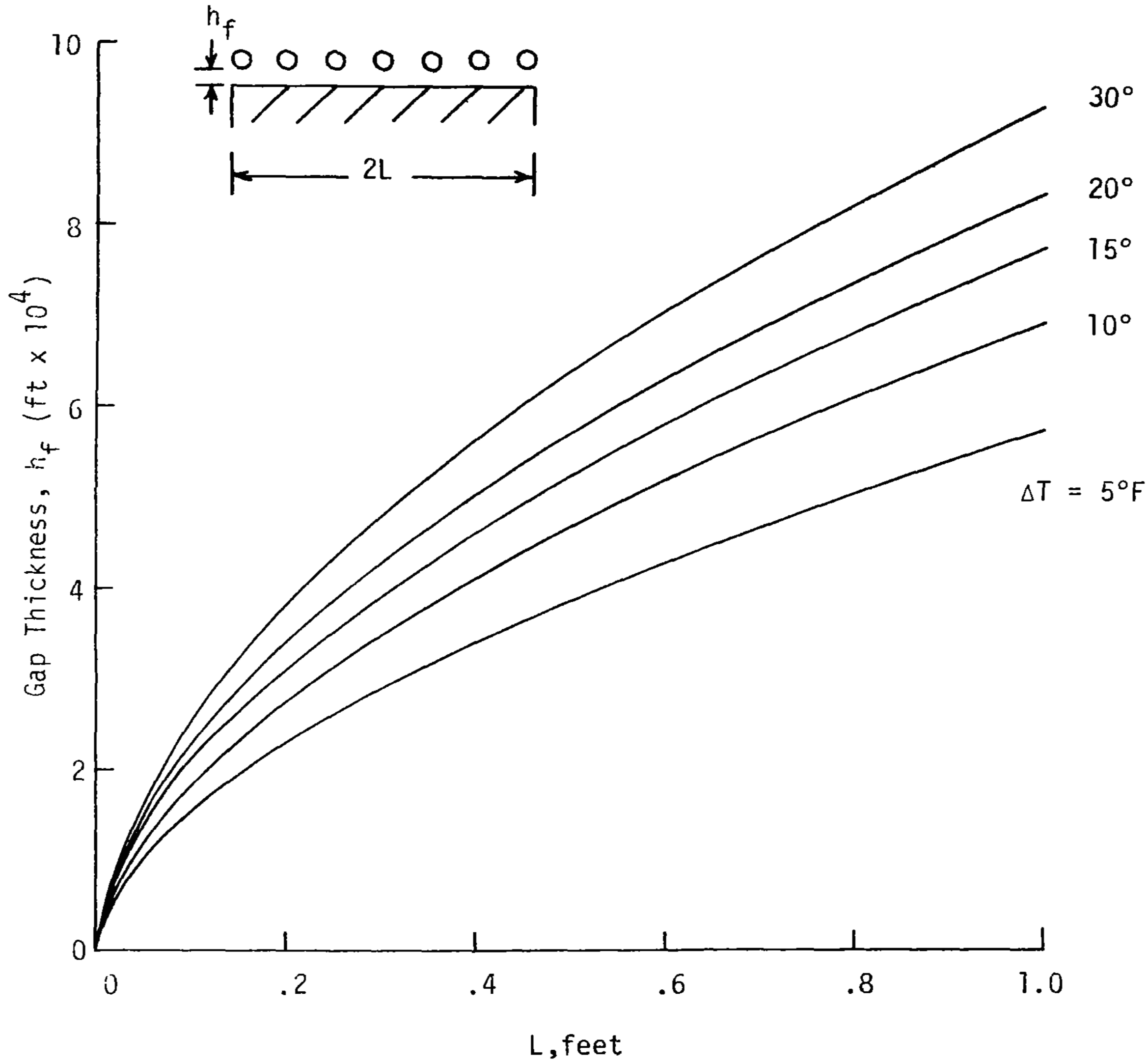

FIGURE A-10. CAPILLARY PUMP GAP THICKNESS VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES

WORKING FLUID: Potassium at 1200°F

Pump Pressure Head = 81.2 lbf/ft^2

Screen Thickness = 4 mils

Total Heat Flow Rate = 5.24×10^6 BTU/hr

Two-Dimensional Analysis

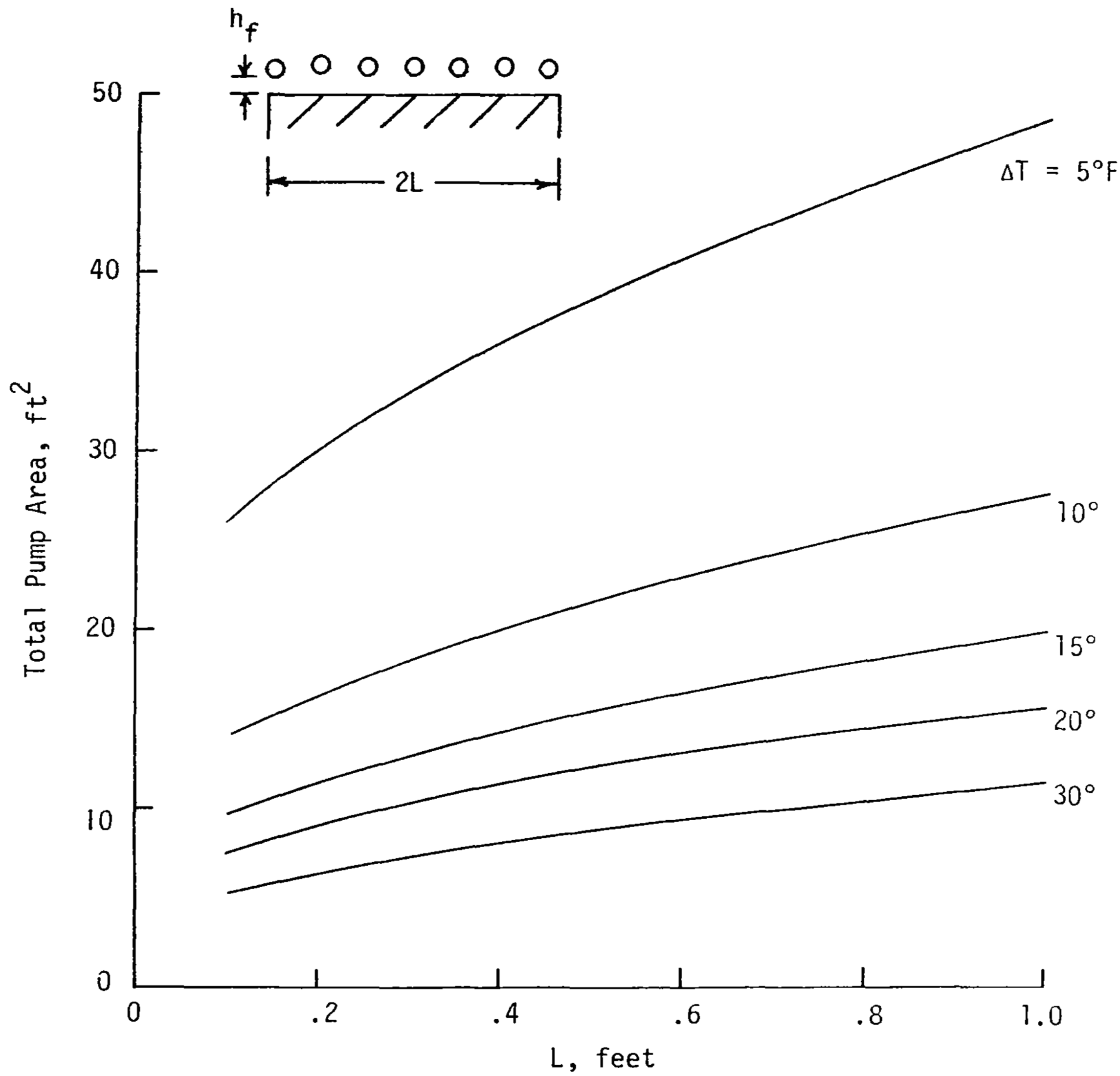

FIGURE A-11. CAPILLARY PUMP AREA VERSUS L FOR VARIOUS TEMPERATURE DIFFERENCES

Thus, for a given working fluid, a maximum velocity implies a maximum value of $\Delta P \Delta T$.

The transition, Reynolds number for pipe flow, is 2000. For a thickness $h = 10^{-3}$ ft. and for Potassium at 1200°F ($P = 46.15$ lbc/ft^3, $\mu = .357$ lbm/hr. ft).

$$R_e = \frac{\rho V h}{\mu} < 2000 \rightarrow V < 4.3 \text{ ft/sec}$$

Then, from equation (a.38)

$$\Delta P \Delta T < 2320 \frac{\text{lbfR}}{\text{ft}^2}$$

or for $\Delta P = 81.2$ lbf/ft^2

$$\Delta T < 29°\text{F}$$

It also turns out for this case that convection terms are not negligible. For a length L, of 1 ft., and $V_2(L) = 4.3$ ft/sec

$$P V_z \frac{\Delta V_z}{\Delta z} = 26.6 \text{ lbf/ft}^2$$

and this is not small compared with 81.2 lbf/ft^2. The other convective term has the same order of magnitude.

The equations which describe the two-dimensional heat pump are summarized as follows:

$$\frac{Q}{Q_o} = \left(\frac{h_f}{h_f + h_s}\right)^{3/4}$$

$$h^4 = h_f^{\,3}(h_f + h_s)$$

$$Q_o = \frac{k \Delta T W L}{h}$$

$$h = \left\{\frac{6 \mu k \Delta T L^2}{\rho h_{fg} \, \Delta P}\right\}^{1/4}$$

The result when the screen thickness, h_s, is neglected is given by Q_0. Figure A-9 presents h_f and A/A_0 for a 4 mil screen thickness. The quantity A/A_0 is the ratio of required area for a finite h_s to the required area for $h_s = 0$ when Q is fixed. Figures A-10 and A-11 present h_f and $A = WL$, respectively versus L and T for the following heat pump specifications:

Working fluid: Potassium at 1200°F

$\Delta P = 81.2 \text{ lbf/ft}^2$

$h_s = 4$ mils

$Q = 5.24 \times 10^6$ BTU/lbm

The analysis is valid only if the convective terms in the z-momentum equation are negligible and if the flow is laminar. Both of these conditions fail if the velocity is high enough. It has been shown that these requirements limit the operating temperature difference to $\Delta T < 30°F$ for a pump with Potassium at 1200°F as the working fluid, with a pressure head of 8.12 psf, and with a gap thickness of 10^{-3} feet.

4. ANALYSIS OF A THREE-DIMENSIONAL CAPILLARY HEAT PUMP

Section 3 presents an analysis of a two-dimensional heat pump for which the only nonzero velocity component is in the z-direction. For velocities in both the x- and z-directions (see Figure A-12), the continuity and momentum equations generalize to

$$\nabla\cdot\bar{V} = -\frac{k\Delta T}{\rho(h_f+h_s)h_f\, h_{fg}} \tag{A.39}$$

$$\nabla P = \mu\frac{\partial^2\bar{V}}{\partial y^2} \tag{A.40}$$

Convection terms are neglected in the momentum equation and gradients of velocity in the x- and z-directions are assumed small compared with the velocity gradient in the y-direction. As discussed in Section 3 , the first assumption may not be good if ΔT is high enough. The second assumption is very good since velocity changes over a distance $\Delta y \sim 10^{-3}$ ft. are comparable with changes in the x- and y-directions over a distance on the order of one foot.

The solution of (A.40) is:

$$\nabla P = -\frac{12\mu\bar{V}}{h_f^2} \tag{A.41}$$

for boundary conditions $V = 0$ at $y = h_f/2$. Taking the divergence of equation (A.41) and using equation (A.39), we obtain

$$\nabla^2 P = \frac{12\mu k\Delta TL}{\rho h_f^3(h_f+h_s)\, h_{fg}} \tag{A.42}$$

The solution of equation (A.42) for a rectangle of sides 2a and 2b with uniform pressure on the boundary is given on p. 171 of Reference 2. For a square of side 2L, the total pressure drop from edge to center is

$$\Delta P = 1.41\,\frac{6\mu k\Delta TL^2}{\rho h_f^3(h_f+h_s)\, h_{fg}} \tag{A.43}$$

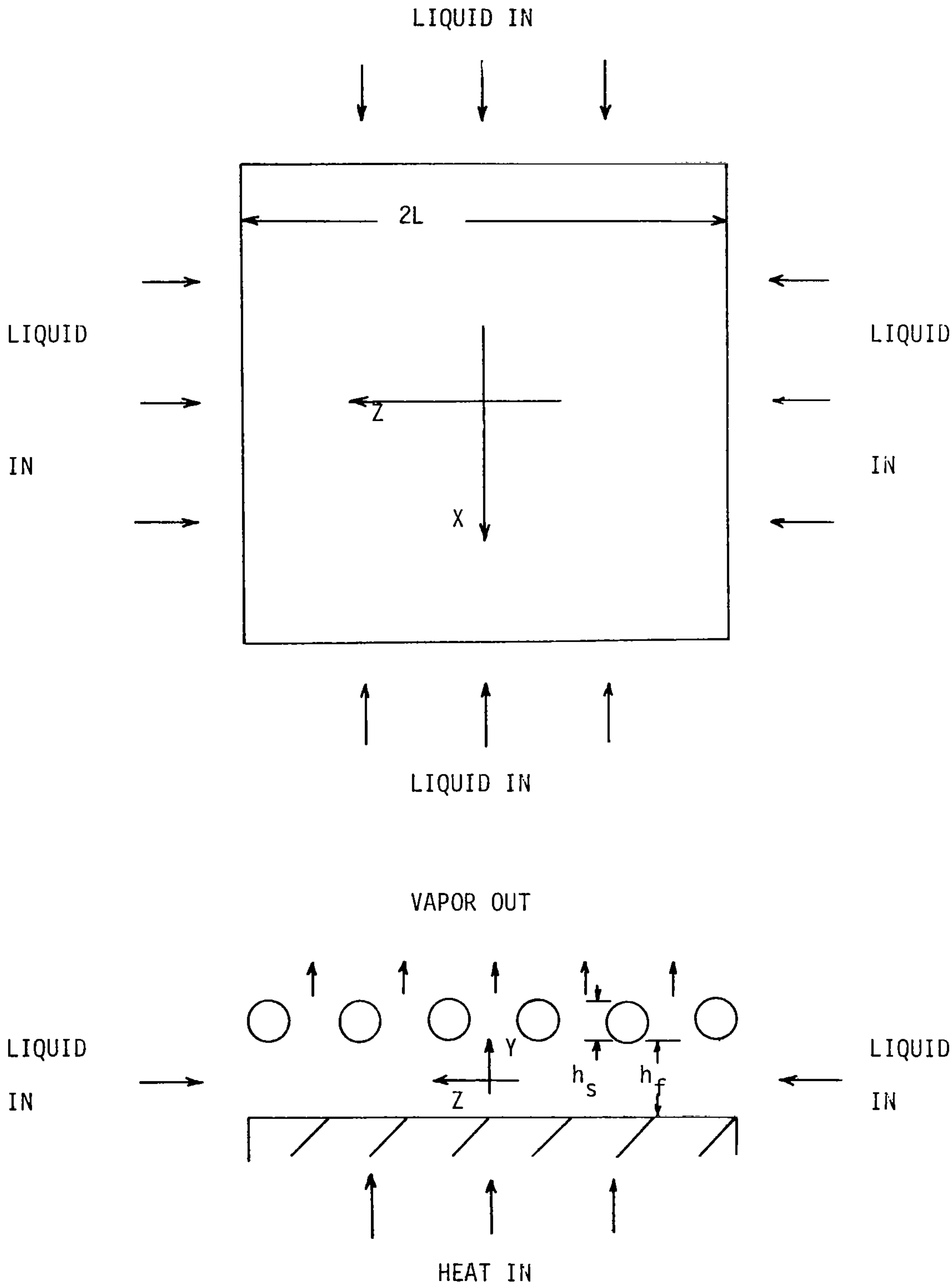

FIGURE A-12. SCHEMATIC OF SQUARE CAPILLARY HEAT PUMP ELEMENT

This is exactly the same result as is obtained for the two-dimensional heat pump (see Section 3 .) except for the constant 1.41. From equation (A.43)

$$h_f^3(h_f+h_s) = \frac{8.46\mu k\Delta TL}{\rho h_{fg} P} \tag{A.44}$$

Defining $Q = \int_0^L (dQ/dz)dz$

where $dQ = qLdz = \frac{k\Delta T}{(h_f+h_s)} Ldz$

we obtain $Q = \frac{k\Delta TL^2}{h_f+h_s}$ (A.45)

For $h_s = 0$

$$h_f \sim \sqrt[4]{1/\Delta P}$$

from equation (A.44). Thus, the total heat flow rate per unit area is greater for the square element than for the two-dimensional element by the factor

$$\sqrt[4]{1.41} = 1.09$$

The quantity h_s appears in the two-dimensional analysis of Section 3 in the same way, and the results for gap thickness and total area in Section 3 . can be used for the square heat pump if appropriately modified. From Figure A-9, it is seen that

$$h_f \text{ versus } h = \sqrt[4]{h_f^3(h_f+h_s)}$$

is very nearly linear for all values of h_f. Thus, h_f for the square pump will be close to nine percent greater. From equation (A.45)

$$A \sim (h_f+h_s)$$

and the ratio of square pump area to the 2 dimensional pump area is

$$\frac{h_f+h_s}{1.09h_f+h_s}$$

Thus, the decrease is substantially less than nine percent for reasonable values of h_s.

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2. Phillips, E. C., "Low Temperature Heat Pipe Research Program," NASA CR-66792, June 1969.

3. Cotter, T. P., "Theory of Heat Pipes," LA-3246-MS, Los Alamos Scientific Lab., 26 March 1965.

4. Carslaw, H. S. and J. C. Jaeger, "Conduction of Heat in Solids," Second Edition, Oxford University Press, London, 1959.

NOMENCLATURE

Symbols

A	=	area
d	=	diameter
f_o	=	2×10^6 1/sec
g	=	acceleration due to gravity
h	=	wick thickness
h_f	=	gap thickness between condenser and bottom of screen
h_{fg}	=	heat of vaporization
h_s	=	screen thickness
K_p	=	wick permeability
k	=	thermal conductivity
L	=	length
m	=	mass flow rate
N	=	number of pumps
P	=	pressure
ΔP	=	pressure difference
Q	=	heat flow rate
Q_o	=	total heat flow rate for $h_s = 0$
R	=	gas constant
Re	=	Reynold's number
r	=	radius
r_c	=	effective pore radius
T	=	temperature
ΔT	=	temperature difference between exterior of cylinder and vapor
$\overline{U}, \overline{V}$,	=	average velocity (volume flow rate per unit area) in axial direction, radial direction
W	=	width
α	=	(minimum fin cross-sectional area)/$2\pi r_2 L$
β	=	(conduction path length)/$(r_v - r_2)$
θ	=	wetting angle, azimuth in cylindrical coordinates
μ	=	viscosity coefficient
ρ	=	density
σ	=	surface tension

Subscripts

ℓ	=	liquid
v	=	vapor
w	=	wick

APPENDIX B

PHYSICAL AND THERMODYNAMIC PROPERTY EQUATIONS

The physical and thermodynamic properties for the three potential working fluids (sodium, potassium, and cesium) were obtained in equation form from References 1 through 7. The specific property data required for the study was:

- o Vapor pressure - temperature
- o Enthalpy of saturated liquid and vapor
- o Heat of vaporization
- o Density of saturated liquid
- o Equation of state for vapor
- o Viscosity of liquid and vapor
- o Specific heat of liquid
- o Surface tension of liquid
- o Thermal conductivity of liquid.

The equations used for calculating these properties are given in this Appendix.

Subroutines were written for the fluid properties, so that they were calculated in the program as a function of fluid and local temperature. The conversion coefficients were included in the individual subroutines, as required, so that the properties given by the program are in the Engineering system of units.

The physical constants for the three fluids are given in Table 1.

Table 1. Physical Constants of Fluids

	Sodium		Potassium		Cesium	
	Engr. Units	cgs Units	Engr. Units	cgs Units	Engr. Units	cgs Units
Atomic Weight	22.991		39.100		132.91	
Melting Temperature °F,°C	208.15	97.86	145.8	63.2	83.5	28.6
Liquid density at the Melting Temp. lb/ft^3, gm/cm^3	57.78	0.9255	51.38	0.8230	114.74	1.838
Heat of Fusion, Btu/lb, cal/gm	48.67	27.04	25.74	14.30	7.043	3.913
Normal Boiling Temp. °F,°C	1620.1	882.3	1400.0	760.0	1236.6	669.2
Heat of Vaporization at the Normal Boiling Temp. Btu/lb, cal/gm	1676.7	931.48	801.63	445.35	212.65	118.14
Critical Temperature °F,°C	4172	2300 ± 60	3452	1900 ± 50	3092	1700 ± 10
Critical Pressure, psia	4953	337 ± 40	2425	165 ± 20	1940	132 ± 8
Critcal Density, lb/ft^3, gm/cm^3	0.371	0.206 ± 0.016	0.364	0.202 ± 0.015	0.781	0.434 ± 0.005

1.1 PROPERTY EQUATIONS FOR SODIUM

- Vapor Pressure of Saturated Liquid

$$\log P\,(\mathrm{atm}) = 3.84235 - \frac{5153.7}{T(°K)} + 0.20171 \log T(°K)$$

- Enthalpy of Saturated Liquid

$$h_{\ell}(\mathrm{cal/gm}) = 16.151 + 0.33994t(°C) - 3.8984\times10^{-5}t^{2}(°C) + 1.4834\times10^{-8}t^{3}(°C)$$

The reference temperature for this equation is 25°C.

- Heat of Vaporization

$$h_{fg}(\mathrm{cal/gm}) = 1157.65\left[1 - \frac{T(°K)}{2573}\right]^{0.3647}$$

- Density of Saturated Liquid

$$\rho_{\ell}(\mathrm{gm/cm^3}) = 0.206 + 1.017\times10^{-2}(2300-t)^{1/2} + 1.1 \times 10^{-4}(2300-t)$$

o <u>Equation of State for Sodium Vapor</u>

$$\frac{Pv}{RT} = 1 + \frac{B}{v} + \frac{C}{v^2} + \frac{D}{v^3} \qquad (T = °K)$$

$$\log B = -2.3013 + \frac{3752.9}{T} + \log T \qquad (cm^3/mol)$$

$$B < 0$$

$$\log C = 2.9771 + \frac{6021.7}{T} \qquad (cm^3/mol)^2$$

$$C > 0$$

$$\log D = 5.2956 + \frac{7521.7}{T} \qquad (cm^3/mol)^3$$

$$D < 0$$

o <u>Viscosity of the Liquid</u>

$$\log \mu(mp) = 1.31558 + \frac{234.00}{T(°K)} - 0.42966 \log T(°K)$$

o <u>Viscosity of the Vapor</u>

$$\mu(\frac{lb}{ft\,hr}) = 0.0275 + 0.175(10^{-4})T(°F)$$

o <u>Specific Heat of the Liquid</u>

$$C_p(cal/gm°C) = 0.33994 - 7.7968 \times 10^{-5} t(°C) + 4.4502 \times 10^{-8} t^2(°C)$$

o <u>Specific Heat of the Vapor</u>

$$C_p(Btu/lb°R) = 0.21598 + 6.053 \exp(-37280/T) \; (°R)$$

o <u>Surface Tension</u>

$$\sigma(\frac{dynes}{cm}) = 202.14 - 0.0986\, T(°C)$$

o <u>Thermal Conductivity of the Liquid Sodium</u>

$$k(\frac{Btu}{hr\,ft\,°F}) = 47.97 - 0.01035T(°F)$$

1.2 PROPERTY EQUATIONS FOR POTASSIUM

o Vapor Pressure of Saturated Liquid

$$\log P\ (\text{atm}) = 2.5324 - \frac{4068.9}{T(°K)} + 0.4665 \log (T°K)$$

o Enthalpy of Saturated Liquid

$H = 9.578 + 0.19042t(°C) - 1.6533 \times 10^{-5}t^2(°C) +$
$2.2490 \times 10^{-8}t^3(°C)$. The reference temperature for this equation is 25°C.

o Density of Saturated Liquid

$$\rho_\ell (\text{gm/cm}^3) = 0.202 + 1.125 \times 10^{-2}\ (1900-t)^{1/2} + 7.601 \times 10^{-5}(1900-t)$$

o Viscosity of the Liquid

$$\log \mu_\ell (\text{mp}) = 2.13880 + \frac{133.039}{T(°K)} - 0.71317 \log T(°K)$$

o Viscosity of the Vapor

$$\mu\left(\frac{\text{lb}}{\text{ft/hr}}\right) = 0.0262 + 0.15(10^{-4})\ T(°F)$$

o Specific Heat of the Liquid

$$Cp_\ell\ (\text{cal/gm °C}) = 0.1904 - 3.3067 \times 10^{-5}t(°C) + 6.7470 \times 10^{-8}t^2(°C)$$

o Specific Heat of the Vapor

$$C_p(\text{Btu/lb°R}) = 0.12700 + 2.888 \exp(-28070/T)\ (°R)$$

o Equation of State

$$P = RT\left(\frac{1}{V} + \frac{B}{V^2} + \frac{C}{V^3} + \frac{D}{V^4} + \frac{E}{V^5} + \frac{F}{V^6}\right) \qquad (T = {}^\circ R)$$

$$B = -0.13019\ T^{0.5}\ \exp\ (7000/T)(cm^3 mole^{-1})$$

$$C = -1.2388 \times 10^7\ \exp\ (900/T)(cm^3 mole^{-1})^2$$

$$D = -1.4179 \times 10^{11}\ \exp\ (1000/T)(cm^3 mole^{-1})^3$$

$$E = 7.9205 \times 10^{15}\ (cm^3 mole^{-1})^4$$

$$F = -3.5099 \times 10^{19}\ (cm^3 mole^{-1})^5$$

o Surface Tension

$$\sigma\ \left(\frac{dynes}{cm}\right) = 121.31 - 0.0739\ T(^\circ C)$$

o Heat of Vaporization

$$h_{fg} = 560.27\left(1 - \frac{T(^\circ K)}{2173}\right)^{0.3558} (cal/g)$$

o Thermal Conductivity of the Liquid Potassium

$$k\left(\frac{Btu}{hr\ ft\ ^\circ F}\right) = 29.15 - 0.008\ T(^\circ F)$$

1.3 PROPERTY EQUATIONS FOR CESIUM

o Vapor Pressure of Saturated Liquid

$$\log P\ (atm) = 3.36292 - \frac{3617.76}{T(^\circ K)} + 0.16005 \log T(^\circ K)$$

o Enthalpy of Saturated Liquid

h_ℓ (cal/gm) = $2.182 + 6.8412 \times 10^{-2} t(^\circ C) - 4.0160 \times 10^{-5} t^2(^\circ C) + 2.6653 \times 10^{-8} t^3(^\circ C)$. The reference temperature for this equation is 25°C.

o Density of Saturated Liquid

$$P_\ell (gm/cm^3) = 0.434 + 2.495 \times 10^{-2}(1770-t)^2 + 2.083 \times 10^{-4}(1770-t)$$

- Viscosity of the Liquid

$$\log \mu_\ell(\text{mp}) = 0.84005 + \frac{205.902}{T(°K)} - 0.27958 \log T(°K)$$

- Viscosity of the Vapor

$$\mu\left(\frac{\text{lb}}{\text{ft hr}}\right) = 0.0337 + 0.25(10^{-4})\ T(°F)$$

- Specific Heat of the Liquid

$$C_{p_\ell}\ (\text{cal/gm°C}) = 0.068412 - 8.032 \times 10^{-5} t(°C) + 7.9959 \times 10^{-8} t^2(°C)$$

- Specific Heat of the Vapor

$$C_p(\text{Btu/lb°R}) = 0.037361 + 1.3099\, \exp(-25663/T)\ (°R)$$

- Equation of State

$$P = RT\left(\frac{1}{V} + \frac{B}{V^2} + \frac{C}{V^3} + \frac{D}{V^4} + \frac{E}{V^5} + \frac{F}{V^6}\right) \qquad (T = °K)$$

$$B = -82.173T^{0.15} \exp\ (3350/T)(\text{cm}^3\text{mole}^{-1})$$

$$C = 2.7318 \times 10^7 \exp\ (1500/T)(\text{cm}^3\text{mole}^{-1})^2$$

$$D = -1.0421 \times 10^{12} \exp\ (550/T)(\text{cm}^3\text{mole}^{-1})^3$$

$$E = 1.2610 \times 10^{16}(\text{cm}^3\text{mole}^{-1})^4$$

$$F = -3.6091 \times 10^{19}(\text{cm}^3\text{mole}^{-1})^5$$

- Surface Tension

$$\sigma\ \left(\frac{\text{dynes}}{\text{cm}}\right) = 76.4 - 0.030[T(°F) - 83]$$

- Heat of Vaporization

$$h_{fg} = 147.12\left(1 - \frac{T(°K)}{2043}\right)^{0.3547} (\text{cal/g})$$

- Thermal Conductivity of Liquid

$$k(\text{Btu/hr ft °F}) = 13.95 - 0.00365T(°F)$$

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APPENDIX C

DESIGN COMPUTER PROGRAM

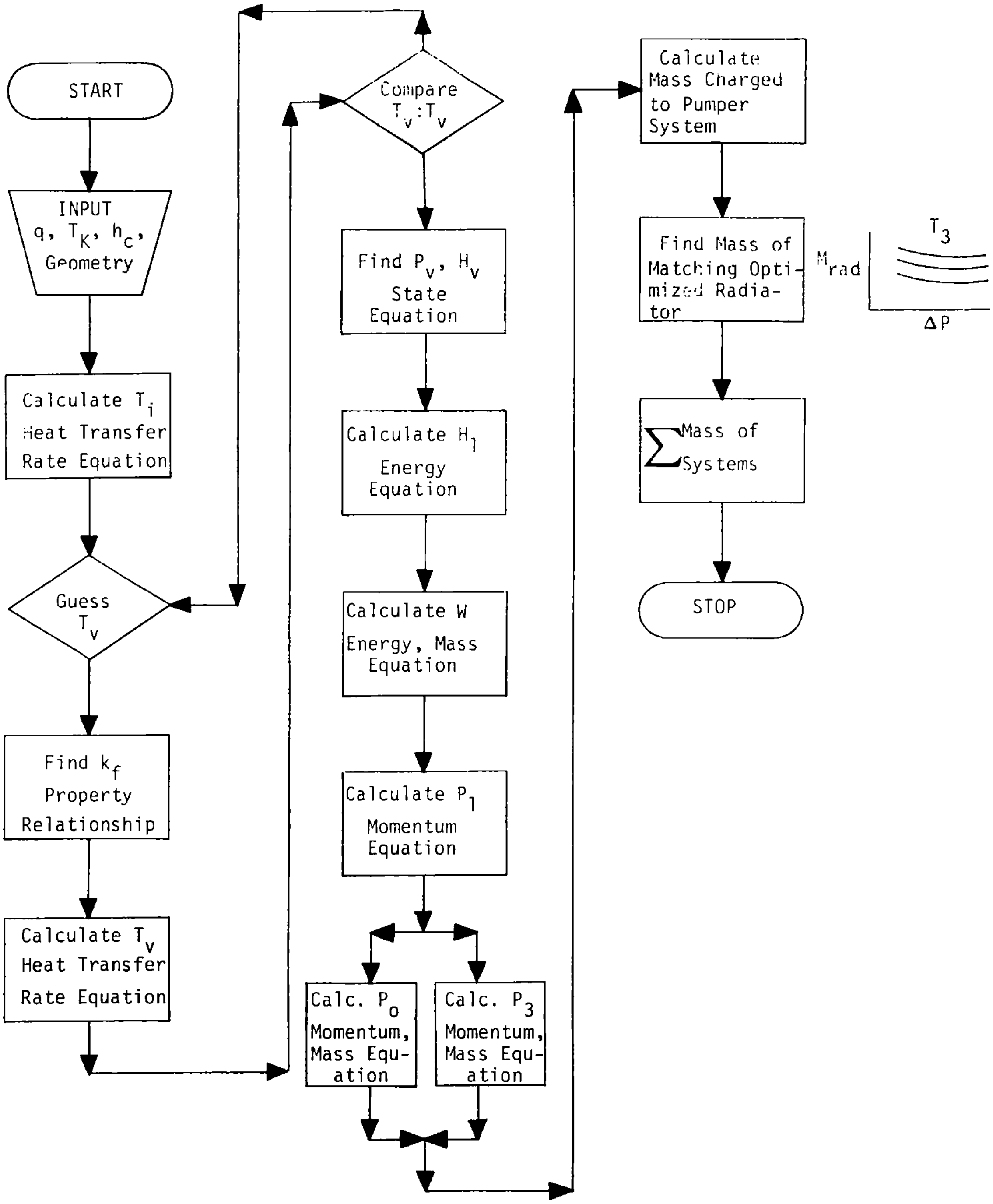
This appendix presents a description of the design computer program. Section 1,Program Abstract, gives a brief summary of the program's purpose and application and shows the calculation sequence. Section 2, Input Data List, gives a definition of the symbols used in the input data block and the format for the data cards. Section 3, Output Data List, gives a similar definition of the data output from the calculations. Section 4, Program Listing, gives a listing of the program card deck and it also gives listings for subroutines used to calculate the properties of the working fluids as a function of temperature.

1. PROGRAM ABSTRACT

The design program was written for the purpose of calculating the optimum radiator size and weight for a specific capillary pump, vapor, and liquid line dimensions. At the same time, the program calculates the pump, working fluid, and line weights and gives a total system weight. Using this program the pump variables can be varied, in turn, and computer runs made to get the corresponding system weights. In this way, the pump dimensions which result in a minimum system weight are obtained. This minimum weight is relative since the heat flux input is arbitrarily set equal to the total heat load divided by the number of heat rejection loops.

The pump dimensions and system variables which were varied, are heat transfer area, wick thickness, pore diameter, distance between liquid feed arteries and the liquid and vapor line diameters and lengths; and heat flux (number of loops), working fluid, and working fluid subcooling.

The following block diagram shows the computing sequence.


START
INPUT
q, T_K, h_c,
Geometry
Calculate T_i
Heat Transfer
Rate Equation
Guess
T_v
Find k_f
Property
Relationship
Calculate T_v
Heat Transfer
Rate Equation
Compare
$T_v : T_v$
Find P_v, H_v
State
Equation
Calculate H_1
Energy
Equation
Calculate W
Energy, Mass
Equation
Calculate P_1
Momentum
Equation
Calc. P_o
Momentum,
Mass Equ-
ation
Calc. P_3
Momentum,
Mass Equ-
ation
Calculate
Mass Charged
to Pumper
System
Find Mass of
Matching Opti-
mized Radia-
tor
M_{rad}
T_3
ΔP
$\sum$ Mass of
Systems
STOP

2. INPUT DATA LIST

The following table describes the individual input card format and defines the symbols used in the data block. Real variables and integer variables are defined according to standard FORTRAN rules (integers begin with i, j, k, l, m, or n).

LIST OF SYMBOLS USED IN DATA BLOCK

SYMBOL	DESCRIPTION	UNITS	LIMIT
<u>CARD 10 (12A6)</u>			
ANOTE	Any comment may be typed in Col. 1-72. This comment will be printed on each run.		
<u>CARD 20 (8E9.4)</u>			
HC	Condensation coefficient	Btu/hr-ft^2-°F	
DXS	Thickness of metal between power conversion loop and cap-pumper.	ft	
XKS	Thermal conductivity of metal between power conversion loop and cap-pumper.	Btu/hr-ft -°F	
TK	Temperature of potassium in power conversion loop.	°F	
<u>CARD 30 (8E9.4)</u>			
QTOT	The total heat to be rejected by capillary-pump loop radiators.	Btu/hr	
A	Area of capillary-pump liquid to vapor interface (screens) plus area of distribution manifold (normal to heat flow)	ft^2	
TV	Temperature at liquid to vapor interface.	°F	
DXF	Thickness of liquid metal in capillary-pump from metal to liquid to vapor interface.	(Btu/hr)ft-°F	

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 40 (8E9.4)			
R	Capillary effective radius	ft	
GEE	Acceleration due to gravity	ft/sec^2	
CARD 50 (8E9.4)			
DTSC	Subcooling at capillary-pump inlet	°F	
CARD 60 (8E9.4)			
XLIQ	Length of tubing from radiator outlet to capillary-pump inlet (liquid line)	ft	
XVAP	Length of tubing from capillary-pump outlet to radiator inlet (vapor line)	ft	
XPIPL	Liquid line - tube wall thickness	ft	
XPIPV	Vapor line - tube wall thickness	ft	
DENM	Density of liquid line and vapor line tubing	lb/ft^3	

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 70 (3E9.4,I5)			
W	Capillary-pump fluid flow rate	lb/hr	
D1	Liquid line ID	ft	
D2	Vapor line ID	ft	
I	Capillary-pump fluid (1=Na, 2=K, 3=Ce)	--	
CARD 80 (2E9.4,4I5)			
ID	Convergence criteria for iteration on temperature at liquid vapor interface.	°F	
WD	Convergence criteria for iteration on flow rate	lb/hr	
NCC	Number of iterations (upper limit) before program aborts.	--	
III	Program control trigger. If III ≤ 0, program will print out intermediate calculated values of selected variables. If III ≥ 1, program will not print out these variables.		
NLOOP	Number of capillary pumped loops		
NUM	Number of elements per capillary-pumped loop		

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 90 (2I5)			
M10	Number of Z10 curves in family of curves (curve 10). Curves provided by D. Wanous. Radiator weight as a function of ΔP for several temperatures at radiator inlet.		12
N10	Number of sets of X10 and Y10 points per curve		12
CARD 100+, 101 etc. (E9.4/8E9.4)	The number of cards depends upon the values of M10 and N10 on Card 90. There will be one card for each M10 plus one card for each of four (X10, Y10) pairs.		
Z10	Identifying temperature (must be input in increasing order) of the curve - where the temperature is at the inlet to the radiator.	°F	12
X10	ΔP across radiator (must be input in increasing order).	psi	12 per ea.Z10
Y10	Total weight of radiator per capillary pump loop.	lbs	12 per ea.Z10
CARD 110 (2I5)			
KAA	Number of areas, A, to be studied.	--	10
CARD 120+, 121 etc. (8E9.4)	The number of cards depends upon the value of KAA. There will be a number of cards equal to the first integer larger than (KAA/8).		

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
AA	Value of area, A, to be studied	ft^2	
CARD 130 (2I5)			
KGAP	Number of gap widths, DXF, to be studied	--	10
CARD 140+, 141 etc. (8E9.4)	The number of cards depends upon the value of KGAP. There will be a number of cards equal to the first interger larger than (KGAP/8).		
GAP	Value of gap, DXF, to be studied	ft	
CARD 150 (2I5)			
KNUM	Number of elements, NUM, to be studied	--	10
CARD 160+, 161 etc. (8I9)	The number of cards depends upon the value of KNUM. There will be a number of cards equal to the first interger larger than (KNUM/8).		
KNUM	Value of number of elements, NUM, studied	--	

3. OUTPUT DATA LIST

The principal output for this program is self-explanitory since it does not use computer symbols. The name of each parameter is printed out in full. This print-out is divided into two general groups: The input configuration and operating parameters are printed out first, and the matched system parameters to which the program converges. This output is arranged according to the following table.

Table 1

Capillary Pump - Program 1 Case No.

Units: lb, ft, hr, Btu, deg F except pressure, psi

Design Runs for (working fluid)	Number Loops	Date

Configuration:

Fluid
Number of Loops
Heat Rate Per Loop
Temperature at Heat Source
Heat Transfer Area
Area
Gap Width
Capillary Eff. Radius
Number of Elements
Element Half Width
Tubing (Liquid)
  ID
  Thickness
  Length
Tubing (Vapor)
  ID
  Thickness
  Length

Matched System Parameters:

Station	1	2	3	4
Flow Rate				
Temperature				
Pressure				
Density				

Table 1 (Cont'd)

Weights/Loop:

Capillary Pump	
Structure	=
Fluid	=
Pump Total	=
Tubing	
Tubing (Liq)	=
Liquid	=
Tubing (Vap)	=
Vapor	=
Tubing Total	=
Radiator	=
System Total	=

In addition to the output listed above, the program user may select (set III = 0 on input Card 8) to have other system state conditions and fluid properties printed out. These appear before the principal output and are arranged as follows:

Table 2

TI	=	Temperature at condenser wall - working fluid interface
J	=	Iteration counter
UF	=	Conductance of the working fluid layer
TVD	=	Difference in the calculated and the guess vapor temperature
PVAP2	=	Vapor pressure in the pump
HVAP2	=	Latent heat of vaporization in the pump
DENV2	=	Density of the vapor in the pump
PL1	-	Static pressure in the liquid line at the entrance to the pump
XMUL1	=	Viscosity of the liquid at the entrance to the pump
XFRIC	=	Frictional pressure loss in the pump
CAP	=	Capillary pressure rise in the wick
PVAP3	=	Static pressure at the entrance to the radiator

<u>Table 2 (Cont'd)</u>

PT3	=	Total pressure at the entrance to the radiator
DELP	=	Pressure drop across the radiator
WTRAD	=	Flow rate in the radiator
SKF	=	Thermal conductivity of the working fluid
SV1	=	First guess of the vapor temperature
TV	=	Vapor temperature
HL2	=	Saturation enthalpy of the vapor
SIG2	=	Surface tension of the working fluid
PT1	=	Dynamic pressure of the liquid
DPLIQ	=	Pressure drop in the liquid line
PRISE	=	Net pressure rise through the pump
DENV3	=	Vapor density at the radiator inlet
WTRADT	=	Total flow rate through all radiators

4. PROGRAM LISTING

```
 I FOR MAIN,MAIN
C CAPILLARY PUMPER PROGRAM 1
C   THIS STEADY STATE PROGRAM COMPUTES THE CONDITIONS                    00110
C   AT THE INLET AND EXIT OF THE RADIATOR                                00120
C
C
      DIMENSION X10(12,12),Y10(12,12),Z10(12),A10(22,12)                 04620
      DIMENSION PVAP(30),DENV(30)                                        01770
      DIMENSION AA(10)                                                   00290
      DIMENSION GAP(10),NNUM(10)
      DIMENSION ANOTE(12)
  999 CONTINUE
C
      READ (5,352) ANOTE
      READ (5,340) HC,DXS,XKS,TK
      READ (5,340) QTOT,A,TV,DXF
      READ (5,340) R,GEE,XMTOT
      READ (5,340) COE,DTSC
      READ (5,340) XLIQ,XVAP,XPIPL,XPIPV,DENM
      READ (5,341) W,D1,D2,I
      READ (5,342) TD,WD,NCC,III,NLOOP,NUM
C     READ IN CURVE 10 HERE
C CURVE 10 IS WANOUS RADIATOR WEIGHT CURVE
      READ (5,343)M10,N10
      DO 22 J=1,M10
   22 READ (5,344)Z10(J),(X10(I10,J),Y10(I10,J),I10=1,N10)
      READ (5,343) KAA
   32 READ (5,340) (AA(II),II=1,KAA)
      READ (5,343)KGAP
   33 READ (5,340)(GAP(IGAP),IGAP=1,KGAP)
      READ (5,343)KNUM
   34 READ (5,351)(NNUM(INUM),INUM=1,KNUM)
C
      CALL DCCDIS (X10,Y10,Z10,A10,M10,N10,FN)
      IF(III-1)83,84,84
   83 WRITE(6,9917)FN
   85 WRITE (6,9918)Z10(1),Z10(2)
      WRITE (6,9918)X10(1,1),Y10(1,1),X10(2,1),Y10(2,1)
      WRITE (6,9918)X10(1,2),Y10(1,2),X10(2,2),Y10(2,2)
   84 CONTINUE
C
      PER=0.9
C THIS IS AN ASSUMED FRACTION OF HEAT TRANSFER TO TOTAL AREA
C
      A1=(3.1417*D1**2)/4.                                               00300
      A2=(3.1417*D2**2)/4.                                               00320
      QUE=QTOT/FLOAT(NLOOP)
      NO=0
C
C DO LOOP FOR AREA VARIATION.
      DO 99 II=1,KAA
C
      A=AA(II)                                                           00350
      AHT=PER*A
```

```
C
C INSERT DO LOOPS HERE FOR GAP AND NUMBER OF ELEMENT VARIATIONS
      DO 99 INUM=1,KNUM
      NUM=NNUM(INUM)
      EA=(A*PER)/FLOAT(NUM)
      XL=SQRT(EA)/2.
      DO 99 IGAP=1,KGAP
      DXF=GAP(IGAP)
C
C NO IS A COUNTER FOR CASE NUMBER.
      NO=NO+1
C
C TI OR TEMPERATURE AT CAP-PUMPER SIDE OF METAL WALL
      TI= TK-QUE*(1./HC+DXS/XKS)/(A*PER)
C
      IF(III-1) 70,71,71
   70 WRITE (6,9901) TI
71    CONTINUE
C
C ITERATION FOR TEMPERATURE AT LIQUID VAPOR INTERFACE (SCREENS) IS PERFORMED
C         HERE.
C ITERATION IS REQUIRED BECAUSE OF THERMAL COND. BEING A FUNCTION OF T
C
      DO 73 J=1,NCC
3     UF=SKF(I,((TI+TV)/2.))/DXF
      TV1 = TI-(QUF/(A*UF*PER))
      TVD = (TV1-TV)
C ITERATION TEST PERFORMED HERE.
      IF (ABS(TV1-TV).LT.TD) GO TO 5
      TV=TV1
      IF(III-1)72,73,73
   72 CONDK=SKF(I,((TI+TV)/2.))
      WRITE (6,9902) J,CONDK
      WRITE (6,9903) UF,TV1
      WRITE (6,9904) TVD,TV
73    CONTINUE
    4 WRITE (6,9905) TVD,J
      STOP
5     TV = TV1
6     CONTINUE
C
C STATION 2 CHARACTERISTICS AS A FUNCTION OF TEMPERATURE COMPUTED HERE.
C
      PVAP2 = PV(I,TV)
      HVAP2 = HSAT(I,TV)+HVX(I,TV)
      HL2 = HSAT(I,TV)
      DENV2=RHOV(I,TV)
      SIG2 = SIG(I,TV)
      IF (III-1) 74,75,75
   74 WRITE (6,9907) PVAP2,TV
      WRITE (6,9908) HVAP2,HL2
      WRITE (6,9909) DENV2,SIG2
75    CONTINUE
      TL1=TV-DTSC
```

```
      HL1 = HSAT(I,TL1)
      SIGL1 = SIG(I,TL1)
      RHOL1=RHOL(I,TL1)
      H12=HSAT(I,TV)-CPL(I,(TV-(DTSC/2.)))*DTSC
      W=QUE/(HSAT(I,TV)+HVX(I,TV)-H12)
      XFRIC=FRIC(A,DXF,TI,TV,DTSC,PER,NUM,I)
C PRESSURE AT STATION 1 COMPUTED HERE.
      PT1=PVAP2+(((1./(DENV2*A2**2))-(1./(RHOL1*
     X    A1**2)))*(W**2)/4.18E+08-2.*SIG2/R)/144.
     X +XFRIC
      CAP=SIG2/(R*72.)
      XMOMLS=((1./(DENV2*A2**2))-(1./(RHOL1*A1**2)))*(W**2/(4.18E+08*144
     X.))
      PL1=PT1-(.5*W**2./(RHOL1*A1**2.*144.*4.18E+08))
      IF(PL1)61,62,62
   61 PL1=0.
      PT1=.5*W**2./(RHOL1*A1**2.*144.*4.18E+08)
   62 CONTINUE
      PRISE=PVAP2-PT1
      FRLOSS=XFRIC/PRISE
      XMUL1=XMUL(I,TL1)
C PIPE LOSSES FROM STA 0 TO 1 COMPUTED HERE.
      DPLIQ=0.263E-11*(XMUL1**0.25)*((W/A2)**1.75)*XLIQ/(RHOL1*D2**1.25)
      PT0=PT1+DPLIQ
      IF (III-1) 77,78,78
   77 WRITE (6,9911) PL1,PT1
      WRITE (6,9912) XMUL1,DPLIQ
      WRITE (6,9920) XFRIC,PRISE
      WRITE (6,9919)CAP,XMOMLS
   78 CONTINUE
C PIPE LOSSES FROM STA 2 TO 3 COMPUTED HERE.
C         SMALL INCREMENTS OF PIPE ARE USED DUE TO DENSITY VARIATION
C         AS A FUNCTION OF PRESSURE.
      NN=10
      PVAP(1)=PVAP2
      DENV(1)=DENV2
      XMUV2=XMUV(I,TV)
      XVAPP=XVAP/FLOAT(NN)
      DO 76 J=1,NN
      DPVAP=0.263E-11*(XMUL2**0.25)*((W/A2)**1.75)*XVAPP/
      (DENV(J)*D2**1.25)
      PVAP(J+1)=PVAP(J)-DPVAP
      DENV(J+1)=RHO(I,TV,PVAP(J+1))
76    CONTINUE
      PT3=PVAP(NN+1)
      DENV3=DENV(NN+1)
      PVAP3=PT3-(.5*W**2./(DENV3*A2**2.*144.*4.18E+08))
      PL0=PT0-(.5*W**2./(RHOL1*A1**2.*144.*4.18E+08))
      DELP=PT3-PT0
      IF(DELP) 91,92,92
   91 WRITE (6,326)NO,DELP
      GO TO 99
   92 CONTINUE
      IF (III-1)79,80,80
```

```
   79 WRITE (6,9913) PVAP3,DENV3
      WRITE (6,9914) PT3,PTO
      WRITE (6,9915) DELP,TV
   80 CONTINUE
C RADIATOR WEIGHT READ FROM CURVE.
   20 CALL DCDIS (A10,X10,Y10,Z10,DELP,WTRADT,TV,M10,N10,FN)
      WTRAD=WTRADT/FLOAT(NLOOP)
      IF(III-1)81,82,82
   81 WRITE(6,9916)WTRAD,WTRADT
      WRITE(6,9917)FN
   82 CONTINUE
   31 CONTINUE
C WEIGHT OF VARIOUS CAP-PUMPER ELEMENTS COMPUTED HERE.
C DISTRIBUTION CHANNELS ARE ASSUMED TO BE 1/2 INCH DEEP
      WTL=A*RHOL(I,(TI+TV)/2.)*(DXF*PER+(.0417*(1.-PER)))
      WTM=A*2.2*DXS*536.
      WTPUMP=WTL+WTM
      D1OD=D1+2.*XPIPL
      A1O=3.1417*D1OD**2/4.
      XSEC1=A1O-A1
      WTPIPL=XSEC1*XLIQ*DENM
      WTLIQ=A1*XLIQ*RHOL1
      D2OD=D2+2.*XPIPV
      A2O=(3.1417*D2OD**2)/4.
      XSEC2=A2O-A2
      WTPIPV=XSEC2*XVAP*DENM
      WTVAP=A2*XVAP*(DENV2+DENV3)/2.
      TUBEWT=WTPIPL+WTLIQ+WTPIPV+WTVAP
      WTOTL=WTPUMP+WTRAD+TUBEWT
      TLO=TL1
      RHOO=RHOL1
      TV3=TV
      WRITE(6,300)NO
      WRITE(6,302)
      WRITE(6,353)ANOTE
      WRITE(6,304)I
      WRITE(6,306)NLOOP
      WRITE(6,308)QUE
      WRITE(6,310)TK
      WRITE(6,327) AHT
      WRITE(6,312)A,DXF,R
      WRITE (6,328)NUM,XL
      WRITE(6,313) D1,XPIPL,XLIQ
      WRITE(6,325) D2,XPIPV,XVAP
      WRITE(6,314)
      WRITE (6,350) FRLOSS
      WRITE(6,316)W,W,W,W,TLO,TL1,TV,TV3,PTO,PT1,PVAP2,PT3
      WRITE(6,318)RHOO,RHOL1,DENV2,DENV3,WTM
      WRITE(6,320)WTL,WTPUMP
      WRITE(6,322)WTPIPL,WTLIQ,WTPIPV,WTVAP
      WRITE(6,324)TUBEWT,WTRAD,WTOTL
   99 CONTINUE
      GO TO 999
  300 FORMAT (1H1,//20X,28HCAPILLARY PUMPER - PROGRAM 1 ,
```

```
     112H    CASE NO. ,I3)
  302 FORMAT (1H0,13X,45HUNITS  LB,FT,HR,BTU,DEG F EXCEPT PRESSURE,PSI)
  304 FORMAT (1H0,/,9X14HCONFIGURATION /16X5HFLUID22X2H= ,I8)
306    FORMAT(16X15HNUMBER OF LOOPS12X2H= ,I8)                        00920
308    FORMAT(16X18HHEAT RATE PER LOOP,9X,2H= ,F9.0)                  00930
310    FORMAT(16X29HTEMPERATURE AT HEAT SOURCE = ,F11.2)              00940
  312 FORMAT (16X,18HAREA              9X2H= F11.2/16X9HGAP WIDTH,
     X         18X2H= ,F15.6/16X21HCAPILLARY EFF. RADIUS,6X2H= ,F15.6)
313    FORMAT(16X15HTUBING (LIQUID)/19X2HID,22X,2H= ,F15.6/
     X                   19X9HTHICKNESS,15X,2H= ,F15.6/19X,6HLENGTH,
     X18X,                   2H= ,F15.6)
  314 FORMAT (1H0,9X26HMATCHED SYSTEM PARAMETERS /13X7HSTATION,
     X         16X1H0,9X,1H1,9X,1H2,9X,1H3)
  316 FORMAT (1H0,15X9HFLOW RATE,7X,F8.2,3(2X,F8.2)/16X,11HTEMPERATURE,
     X         5XF8.2,3(2X,F8.2)/16X8HPRESSURE,8XF8.2,3(2X,F8.2))
  318 FORMAT (1H ,15X7HDENSITY,9X,F8.2,3(2X,F8.2)/10X13HWEIGHTS
     X         /,13X16HCAPILLARY PUMPER/16X9HSTRUCTURE4X2H= ,F9.2)
320    FORMAT(16X5HFLUID8X2H= ,F9.2/14X12HPUMPER TOTAL3X2H= ,
     XF9.2)
322    FORMAT(13X6HTUBING/16X12HTUBING (LIQ)3X2H= ,F9.2/
     X              16X6HLIQUID,9X,2H= ,F9.2/16X12HTUBING (VAP)
     X                 2H= ,F9.2/16X5HVAPOR,10X,2H= ,F9.2)
324    FORMAT (14X12HTUBING TOTAL,6X,2H= ,F9.2/13X,8HRADIATOR
     X,                    13X2H= ,F9.2/13X12HSYSTEM TOTAL,12X2H=
     X,F9.2///)
325    FORMAT(16X,14HTUBING (VAPOR)/19X2HID,22X,2H= ,F15.6/
     X                   19X9HTHICKNESS,15X,2H= ,F15.6/19X,6HLENGTH,
     X18X,                   2H= ,F15.6)
  326 FORMAT (1H ,39HCAPILLARY PUMPER WILL NOT PUMP. CASE NO.
     (20H    DELTA PRESSURE = ,F13.4)
  327 FORMAT (16X,18HHEAT TRANSFER AREA9X2H= F11.2)
  328 FORMAT (16X,18HNUMBER OF ELEMENTS,9X,2H= ,I8,/
    X    16X,18HELEMENT HALF WIDTH,9X,2H= ,F15.6  )
  340 FORMAT (8F9.4)
  341 FORMAT (3F9.4,I5)
  342 FORMAT (2F9.4,4I5)
  343 FORMAT (2I5)
  344 FORMAT (F9.4/(8F9.4))
  345 FORMAT (1H ,29HI EXCEEDS PROGRAM LIMITS  I= ,I5)
  350 FORMAT (1H+,70X,11HFRICTION = ,F7.5,23H  OF PUMP PRESSURE   )
  351 FORMAT (8I9)
  352 FORMAT (12A6)
  353 FORMAT (1H ,12A6)
 9901 FORMAT (1H1,7H   TI= ,F13.5)
 9902 FORMAT (1H ,7H    J= ,I13,  8H   SKF= ,F13.5)
 9903 FORMAT (1H ,7H   UF= ,F13.5,8H   TV1= ,F13.5)
 9904 FORMAT (1H ,7H  TVD= ,F13.5,8H    TV= ,F13.5)
 9905 FORMAT (1H ,19HTEMPERATURE DIFF = ,F13.5,7H    J= ,I5)
 9907 FORMAT (1H ,7HPVAP2= ,F13.5,8H    TV= ,F13.5)
 9908 FORMAT (1H ,7HHVAP2= ,F13.5,8H   HL2= ,F13.5)
 9909 FORMAT (1H ,7HDENV2= ,F13.5,8H  SIG2= ,F13.5)
 9911 FORMAT (1H ,7H  PL1= ,F13.5,8H   PT1= ,F13.5)
 9912 FORMAT (1H ,7HXMUL1= ,F13.5,8H DPLIQ= ,F13.5)
 9913 FORMAT (1H ,7HPVAP3= ,F13.5,8H DENV3= ,F13.5)
```

```
 9914 FORMAT (1H ,7H  PT3= ,F13.5,8H    PTO= ,F13.5)
 9915 FORMAT (1H ,7H DELP= ,F13.5,8H     TV= ,F13.5)
 9916 FORMAT (1H ,7HWTRAD= ,F13.5,8HWTRADT= ,F13.5)
 9917 FORMAT (1H ,7H   FN= ,F13.5)
 9918 FORMAT(1H ,8F10.3)
 9919 FORMAT (1H ,7H  CAP= ,F13.5,8HXMOMLS= ,F13.5)
 9920 FORMAT (1H ,7HXFRIC= ,F13.5,8H PRISE= ,F13.5)
      END
```

```
 I   FOR FRIC,FRIC
      FUNCTION FRIC(A,DXF,TI,TV,DTSC,PER,NUM,I)
C THIS FUNCTION COMPUTES PRESSURE LOSS OF LIQUID FLOWING IN CAP-PUMPER
C (PSI) WHERE  A=TOTAL HEAT TRANSFER AREA (SQ FT)
C              DXF=GAP WIDTH (FT)
C              TI=TEMPERATURE AT CONDENSER WALL (DEG F)
C              TV=TEMPERATURE AT SCREENS (DEG-F)
C              DTSC=TEMP DIFF BETWEEN SCREENS AND LIQ IN (DEG-F)
C              PER=FRACTION OF TOTAL HEAT TRANSFER AREA COVERED
C                  BY SCREENS
C              NUM=NUMBER OF CELLS
C              I=TYPE FLUID (1=NA,2=K,3=CE)
      EA=(A*PER)/FLOAT(NUM)
      XL=SQRT(EA)/2.
      TAVG=(TI+TV)/2.
      VIS=XMUL(I,TAVG)
      XK=SKF(I,TAVG)
      H=HVX(I,TV)
      D=RHOL(I,TAVG)
C TWO LAYER SCREEN THICKNESS IS 4 MILS
      DP=(2.02E-08)*VIS*XK*(TI-TV)*XL**2/
     X (D*(DXF-0.000333)**3.*DXF*H)
C 1.05 IS A FACTOR TO ACCOUNT FOR LOSSES IN THE DISTRIBUTION MANIFOLD
      FRIC=(              DP*1.05)/144.
      RETURN
      END
```

```
 I   FOR RHO,RHO
      FUNCTION RHO(I,XT,XP)
C THIS FUNCTION COMPUTES DENSITY BASED ON EQ OF STATE
C LB/CU FT WHERE  (I) IS NA=1,K=2,CE=3 XT=DEG F,
C XP=LB/SQ-IN
      DIMENSION B(3),R(3)
      P=XP*144.
      T=XT+460.
      TK=T/1.8
      B1=-2.3013+(3752.9/TK)+ALOG10(TK)
      B(2)=-0.13019**SQRT(TK)*EXP(7000./TK)*3.531E-05/.0861
      B(1)=10.**ABS(B1)*3.531E-05/0.0503
```

```
      B(3)=(-82.173*T**.15)*EXP(3350./TK)*3.531E-05/.293
      R(1)=1545./22.991
      R(2)=1545./39.1
      R(3)=1545./132.91
      AA=B(I)*R(I)*T
      BB=R(I)*T
      RHO=2.*(-P)/(-BB-SQRT(BB**2.-4.*AA*(-P)))
      RETURN
      END
```

```
 I   FOR RHOV,RHOV
      FUNCTION RHOV(I,T)
C    THIS FUNCTION FINDS VALUES FOR DENSITY OF SAT VAPOR
C     LB/CU FT   WHERE   I(NA=1, K=2, CE=3), T=DEG F
      IF(I.EQ.1) GO TO 10
      IF(I.EQ.2) GO TO 20
      IF(I.EQ.3) GO TO 30
      WRITE (6,9901)
9901  FORMAT(1H 'I EXCEEDS PROGRAM LIMIT ')
      STOP
10    IF(T-840.) 40,50,50
40    WRITE (6,9902)T
 9902 FORMAT (1H ,28H T IS BEYOND CURVE LIMITS T=     ,F13.4)
      STOP
50    IF(T-1300.) 55,55,60
55    SL=ALOG(4.E-05/2.6E-03)/ALOG(840./1300.)
      C=ALOG(2.6E-03)-SL*ALOG(1300.)
      RHOV=(T**SL)*EXP(C)
      RETURN
60    IF(T-2120.) 65,65,40
65    SL=ALOG(2.6E-03/.10)/ALOG(1300./2120.)
      C=ALOG(.10)-SL*ALOG(2120.)
      RHOV=(T**SL)*EXP(C)
      RETURN
20    IF(T-840.) 40,70,70
70    IF(T-1050.) 75,75,80
75    SL=ALOG(5.4E-04/4.E-03)/ALOG(840./1050.)
      C=ALOG(4.E-03)-SL*ALOG(1050.)
      RHOV=(T**SL)*EXP(C)
      RETURN
80    IF(T-1550.) 85,85,90
85    SL=ALOG(4.E-03/6.6E-02)/ALOG(1050./1550.)
      C=ALOG(6.6E-02)-SL*ALOG(1550.)
      RHOV=(T**SL)*EXP(C)
      RETURN
90    IF(T-2240.) 95,95,40
95    SL=ALOG(6.6E-02/.41)/ALOG(1550./2240.)
      C=ALOG(.41)-SL*ALOG(2240.)
      RHOV=(T**SL)*EXP(C)
      RETURN
30    IF(T-500.) 40,100,100
```

```
100    IF(T-1000.) 105,105,110
105    SL=ALOG(2.6E-04/2.8E-02)/ALOG(500./1000.)
       C=ALOG(2.8E-02)-SL*ALOG(1000.)
       RHOV=(T**SL)*EXP(C)
       RETURN
110    IF(T-1500.) 115,115,120
115    SL=ALOG(2.8E-02/.3)/ALOG(1000./1500.)
       C=ALOG(.3)-SL*ALOG(1500.)
       RHOV=(T**SL)*EXP(C)
       RETURN
120    IF(T-2240.) 125,125,40
125    SL=ALOG(.3/1.7)/ALOG(1500./2240.)
       C=ALOG(1.7)-SL*ALOG(2240.)
       RHOV=(T**SL)*EXP(C)
       RETURN
       END
```

```
 I   FOR RHOL,RHOL
      FUNCTION RHOL(I,XT)
C THIS FUNCTION COMPUTES DENSITY OF SATURATED LIQUID (LB/CU-FT)
C       WHERE   I(NA = 1, K = 2, CE = 3), XT = DEG F
      T = (XT-32.)/1.8
      IF (I.EQ.1) GO TO 10
      IF (I.EQ.2) GO TO 20
      IF (I.EQ.3) GO TO 30
      WRITE (6,9901)
 9901 FORMAT (1H ,16H I EXCEEDS LIMIT )
      STOP
10    RHOL = (.206+.01017*SQRT(2300.-T)+1.1E-04*(2300.-T))
     X*62.43
      RETURN
20    RHOL = (.202+.01125*SQRT(1900.-T)+7.601E-05*(1900.-T
     X))*62.43
      RETURN
30    RHOL = (.434+.02495*(1770.-T)**.5+2.083E-04*(1770.-T
     X))*62.43
      RETURN
      END
```

```
 I   FOR PV,PV
      FUNCTION PV(I,XT)
C THIS FUNCTION COMPUTES VAPOR PRESSURE OF SAT LIQ (PSI)
C       WHERE   I(NA = 1, K = 2, CE = 3), XT = DEG F
      DIMENSION A(3),B(3),C(3)
      A(1) = 3.84235
      A(2) = 2.5324
      A(3) = 3.36292
      B(1) = 5153.7
```

```
      B(2) = 4068.9                                                     03350
      B(3) = 3617.76                                                    03360
      C(1) = 0.20171                                                    03370
      C(2) = 0.4665                                                     03380
      C(3) = 0.16005                                                    03390
      T = ((XT-32.)/1.8)+273.                                           03400
      PV = (10.**(A(I)-(B(I)/T)+C(I)*ALOG10(T)))*14.7                   03410
      RETURN                                                            03420
      END                                                               03430


 I  FOR HVX,HVX
      FUNCTION HVX(I,XT)                                                03440
C THIS FUNCTION COMPUTES THE HEAT OF VAPORIZATION (BTU/LB)
C       WHERE   I(NA = 1, K = 2, CE = 3), XT = DEG F                    03470
      DIMENSION A(3),B(3),C(3)                                          03480
      A(1) = 1157.65                                                    03490
      A(2) = 560.27                                                     03500
      A(3) = 147.12                                                     03510
      B(1)=2573.                                                        03520
      B(2)=2173.                                                        03530
      B(3)=2043.                                                        03540
      C(1) = 0.3647
      C(2) = 0.3558                                                     03560
      C(3) = 0.3547                                                     03570
      T = ((XT-32.)/1.8)+273.                                           03580
      HVX = (A(I)*(1-T/B(I))**C(I))*1.79882                             03590
      RETURN                                                            03600
      END                                                               03610


 I  FOR HSAT,HSAT
      FUNCTION HSAT(I,XT)                                               03050
C THIS FUNCTION COMPUTES ENTHALPY OF SATURATED LIQUID (BTU/LB)
C       WHERE   I(NA = 1, K = 2, CE = 3), XT = DEG F                    03080
      T = (XT-32.)/1.8                                                  03090
      IF (I-1) 10,10,20                                                 03100
10    HSAT = (16.151+0.33994*T-.38984E-04*T**2.+.14834E-07              03110
     X*T**3.)*                       1.79882                            03120
      RETURN                                                            03130
20    IF (I-2) 30,30,40                                                 03140
30    HSAT = (9.578+0.19042*T-.16533E-04*T**2.+.22490E-07*              03150
     XT**3.)*                        1.79882                            03160
      RETURN                                                            03170
40    IF (I-3) 50,50,60                                                 03180
50    HSAT = (2.182+.06841*T-.40160E-04*T**2.+.26653E-07*T              03190
     X**3.)*                         1.79882                            03200
      RETURN                                                            03210
99    FORMAT(1H 'I EXCEEDS LIMIT ')                                     03230
60    WRITE (6,99  )                                                    03220
```

```
      STOP                                                              03240
      END                                                               03250


 I   FOR CPL,CPL
      FUNCTION CPL(I,XT)                                                02330
C     THIS FUNCTION COMPUTES SPECIFIC HEAT OF LIQUID                    02340
C       (BTU/LB-DEG F) WHERE    I(NA = 1, K = 2, CE = 3)                02350
C        XT = DEG. F                                                    02370
      DIMENSION A(3),B(3),C(3)                                          02380
      A(1) = 0.33994                                                    02390
      A(2) = 0.1904                                                     02400
      A(3) = 0.068412
      B(1) = 7.7968E-05                                                 02420
      B(2) = 3.3067E-05                                                 02430
      B(3) = 8.032E-05                                                  02440
      C(1) = 4.4502E-08                                                 02450
      C(2) = 6.747E-08                                                  02460
      C(3) = 7.9959E-08                                                 02470
      T = (XT-32.)/1.8                                                  02480
      CPL = A(I)-B(I)*T+C(I)*T**2                                       02490
      RETURN                                                            02500
      END                                                               02510


 I   FOR SKF,SKF
      FUNCTION SKF(I,T)
C THIS FUNCTION COMPUTES THE THERMAL CONDUCTIVITY OF LIQUID
C          BTU-FT/HR-SQ FT-DEG F
C          WHERE  I(NA=1,K=2,CE=3), T=DEG F
      DIMENSION A(3),B(3),X11(12),Y11(12),A11(22,4)
      A(1)=47.97
      B(1)=-0.01035
      A(2)=29.15
      B(2)=-0.008
      IF (I-2) 10,10,20
   10 SKF=A(I)+B(I)*T
      RETURN
   20 DATA (X11(J),Y11(J),J=1,6)/200.,12.1,400.,12.0,550.,11.75,
     X 700.,11.4,1150.,9.75,2200.,5.5/
      CALL CCDIS (X11,Y11,A11,6,FN)
      CALL CDIS (A11,X11,Y11,T,SKF,6,FN)
      RETURN
      END


 I   FOR SIG,SIG
      FUNCTION SIG(I,XT)                                                03620
```

```
C     THIS FUNCTION COMPUTES SURFACE TENSION LB/FT
C       WHERE   I(NA = 1, K = 2, CE = 3), XT = DEG F
      DIMENSION A(3),B(3),T(3)
      A(1) = 202.14
      A(2) = 121.31
      A(3) = 76.4
      B(1) = 0.0986
      B(2) = 0.0739
      B(3) = 0.030
      T(1) = (XT-32.)/1.8
      T(2)=(XT-32.)/1.8
      T(3) = XT-83.
      SIG = (A(I)-B(I)*T(I))*(2.248E-06/.03281)
      RETURN
      END
```

```
 I   FOR XMUV,XMUV
      FUNCTION XMUV(I,T)
C THIS FUNCTION COMPUTES VISCOSITY OF VAPOR (LB/HR-FT)
C       WHERE   I(NA = 1, K = 2, CE = 3), T = DEG F
      DIMENSION A(3),B(3)
      A(1) = 0.0275
      A(2) = 0.0262
      A(3) = 0.0337
      B(1) = .175E-04
      B(2) = .15E-04
      B(3) = .25E-04
      XMUV = A(I)+B(I)*T
      RETURN
      END
```

```
 I   FOR XMUL,XMUL
      FUNCTION XMUL(I,XT)
C THIS FUNCTION COMPUTES VISCOSITY OF LIQUID (LB/HR-FT)
C       WHERE   I(NA = 1, K = 2, CE = 3), XT = DEG F
      DIMENSION A(3), B(3), C(3)
      A(1) = 1.31558
      A(2) = 2.13880
      A(3) = 0.84005
      B(1) = 234.
      B(2) = 133.039
      B(3) = 205.902
      C(1) = 0.42966
      C(2) = 0.71317
      C(3) = 0.27958
      T = ((XT-32.)/1.8)+273.
      XMUL=(10.**(A(I)+(B(I)/T)-C(I)*ALOG10(T)))*.242
      RETURN
```

```
      END                                                               02830

 I   FOR TDSEQ,TDSEQ
      SUBROUTINE TDSEQ (A,N,FN)                                         05560
C TDSEQ                                                                 05530
C TDSEQ SOLVES A SYSTEM OF SIMULTANEOUS EQUATIONS                       05540
C WHOSE COEFFICIENT MATRIX IS TRI-DIAGONAL                              05550
      DIMENSION A(22,4)
      IF(A(1,2)) 12,11,12                                               05580
11    FN=1.0                                                            05590
      GO TO 50                                                          05600
12    A(1,3)=A(1,3)/A(1,2)                                              05610
      A(1,1)=A(1,4)/A(1,2)                                              05620
      N= N                                                              05630
      DO 10 I=2,N                                                       05640
      A(I,2)=A(I,2)-A(I,1)*A(I-1,3)                                     05650
      IF(A(I,2)) 13,11,13                                               05660
13    A(I,3)=A(I,3)/A(I,2)                                              05670
10    A(I,1)=(A(I,4)-A(I,1)*A(I-1,1))/A(I,2)                            05680
      I = N                                                             05690
20    I = I-1                                                           05700
      IF(I) 40,40,30                                                    05710
30    A(I,1)=A(I,1)-A(I,3)*A(I+1,1)                                     05720
      GO TO 20                                                          05730
40    FN = 0.0                                                          05740
50    RETURN                                                            05750
      END                                                               05760

 I   FOR CCDIS,CCDIS
      SUBROUTINE CCDIS (X,Y,A,N,FN)                                     05150
C CONTINUOUS DERIVATIVE INTERPOLATION SUBROUTINES                       05140
C CCDIS COMPUTES THE COEFS FOR THE CUBIC EQUATION                       05160
      DIMENSION X(12),Y(12),A(22,4)
      A(1,1)=0.0                                                        05180
      A(1,2)=-2.0                                                       05190
      A(1,3)=2.0*(X(2)-X(1))                                            05200
      A(1,4)=0.0                                                        05210
      N3= N-1                                                           05220
      N1= N3-1                                                          05230
      J= 1                                                              05240
      DO 10 I = 1,N1                                                    05250
      J = J+1                                                           05260
      A(J,1)=(X(I+1)-X(I))/(X(I+2)-X(I+1))                              05270
      A(J,2)=A(J,1)*(X(I+1)-X(I))                                       05280
      A(J,3)=1.0                                                        05290
      A(J,4)=(Y(I+1)-Y(I))/((X(I+1)-X(I))*(X(I+2)-X(I+1)))              05300
      A(J,4)=A(J,4)-(Y(I+2)-Y(I+1))/((X(I+2)-X(I+1))*(X(I+              05310
     X2)-X(I+1)))                                                       05320
```

```
      K = I
      J = J+1
      A(J,1)=(X(K+1)-X(K))/(X(K+2)-X(K+1))
      A(J,2)=-((X(K+2)-X(K))/((X(K+2)-X(K+1))*(X(K+1)-X(K))
     X))
      A(J,3)=1.0
      A(J,4)=(Y(K+2)-Y(K+1))/((X(K+1)-X(K))*(X(K+2)-X(K+1))
     X)
      A(J,4)=A(J,4)/(X(K+2)-X(K+1))
      R=(Y(K+1)-Y(K))/((X(K+1)-X(K))*(X(K+1)-X(K)))
      R=R/(X(K+2)-X(K+1))
10    A(J,4)=A(J,4)-R
      N2=2*N3
      A(N2,1)=-2.0
      A(N2,2)=-4.0*(X(N3+1)-X(N3))
      A(N2,3)=0.0
      A(N2,4)=0.0
      CALL TDSEQ (A,N2,FN)
      RETURN
      END


 I  FOR CDIS,CDIS
      SUBROUTINE CDIS (A,X,Y,XP,YP,N,FN)
C CDIS INTERPOLATES BETWEEN TAB POINTS
      DIMENSION X(12),Y(12),A(22,4)
      IF (X(1)-XP) 11,11,10
10    SLOPE1=(Y(2)-Y(1))/(X(2)-X(1))+A(1,1)*(X(2)-X(1))
      YP=Y(1)+SLOPE1*(XP-X(1))
      FN=-1.0
      GO TO 16
11    N=N
      IF (XP-X(N)) 13,13,12
12    N2=2*(N-1)
      SLOPE2=(Y(N)-Y(N-1))/(X(N)-X(N-1))-A(N2-1,1)*(X(N)
     X    -X(N-1))-A(N2,1)*(X(N)-X(N-1))*(X(N)-X(N-1))
      YP=Y(N)+SLOPE2*(XP-X(N))
      FN=1.0
      GO TO 16
13    I=1
14    I=I+1
      IF (X(I)-XP) 14,15,15
15    K=2*I-3
      C=Y(I-1)+((Y(I)-Y(I-1))/(X(I)-X(I-1)))*(XP-X(I-1))
      C=C+A(K,1)*(XP-X(I-1))*(X(I)-XP)
      YP=C+A(K+1,1)*(XP-X(I-1))*(XP-X(I-1))*(X(I)-XP)
      FN=0.0
16    RETURN
      END
```

```
I   FOR DCCDIS,DCCDIS
      SUBROUTINE DCCDIS (XX,YY,ZZ,B,M,N,FN)
      DIMENSION B(22,12),XX(12,12),YY(12,12),ZZ(12)
      DIMENSION A(22,4),X(12),Y(12)
      L=2*N-2
      DO 13 I=1,M
      DO 10 J=1,N
      X(J)=XX(J,I)
10    Y(J)=YY(J,I)
      CALL CCDIS (X,Y,A,N,FN)
      DO 12 K=1,L
12    B(K,I)=A(K,1)
13    CONTINUE
      RETURN
      END
```

```
I   FOR DCDIS,DCDIS
      SUBROUTINE DCDIS (B,XX,YY,ZZ,XP,YP,ZP,M,N,FN)
      DIMENSION A(22,4),X(12),Y(12)
      DIMENSION B(22,12),XX(12,12),YY(12,12),ZZ(12)
      L=2*N-2
      DO 11 I=1,M
      DO 9 K=1,L
9     A(K,1)=B(K,I)
      DO 10 J=1,N
      X(J)=XX(J,I)
10    Y(J)=YY(J,I)
      CALL CDIS(A,X,Y,XP,A(I,2),N,FN)
11    CONTINUE
      DO 12 K=1,M
12    Y(K)=A(K,2)
      CALL CCDIS (ZZ,Y,A,M,FN)
      CALL CDIS (A,ZZ,Y,ZP,YP,M,FN)
      RETURN
      END
```

APPENDIX D

PERFORMANCE COMPUTER PROGRAM

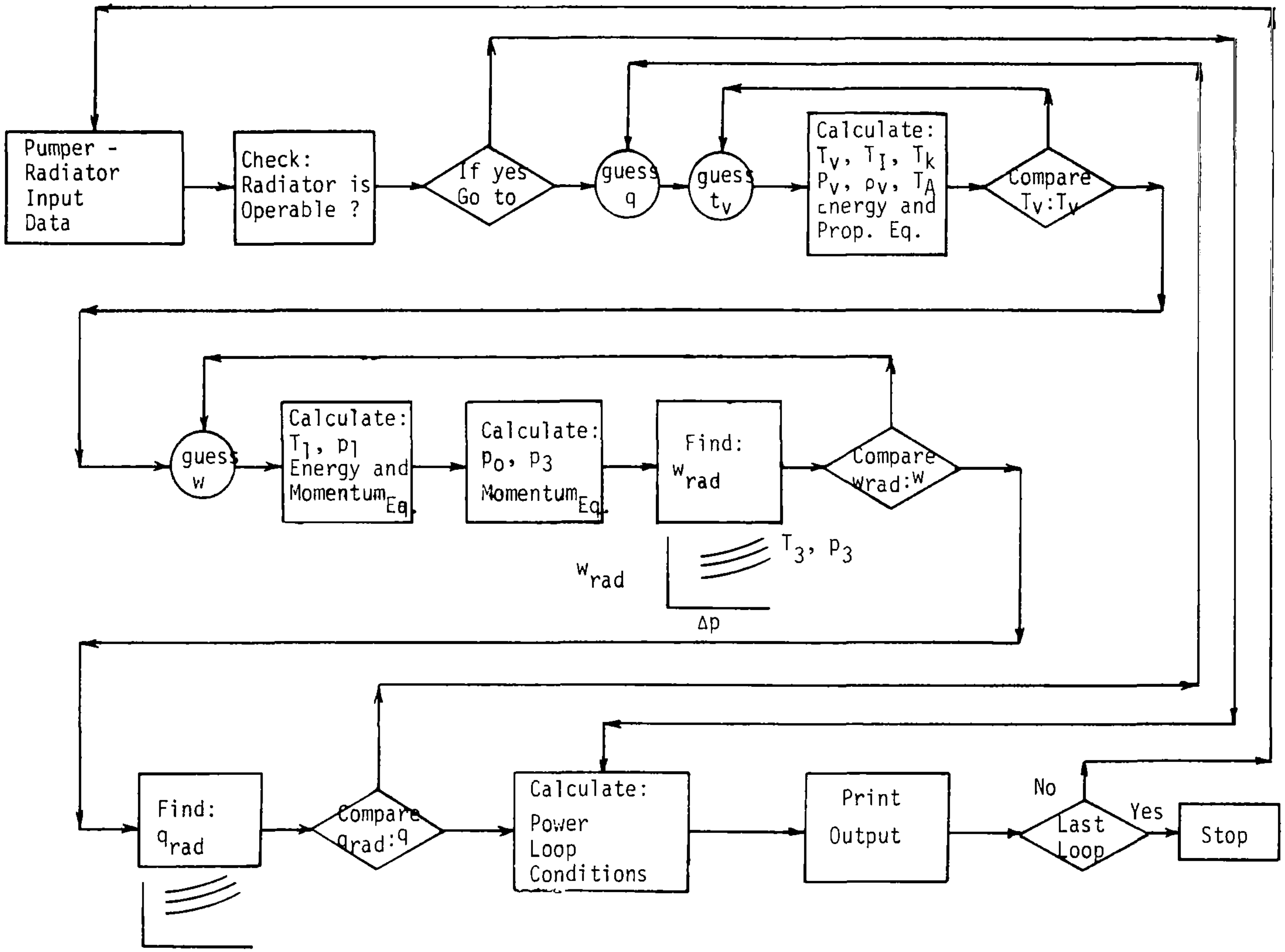
This Appendix presents a description of the performance computer program. Section 1, Program Abstract, gives a brief summary of the program's purpose and application and shows the calculation sequence. Section 2, Input Data List, gives a definition of the symbols used in the input data block and the format for the data cards. Section 3, Output Data List, gives a similar definition of the data output from the calculations. Section 4, Program Listing, gives a listing of the main computer program. The complete program must also be supported with subroutines as follows: RADP, described in Appendix F, for calculating the radiator performance, FRIC, listed in Appendix C, to calculate the liquid friction in the wick and the fluid property subroutines listed in Appendix C. Section 5, An Example Case, gives the input data and output results for a sample case.

1. PERFORMANCE PROGRAM ABSTRACT

The performance program was written for the purpose of calculating the performance of a series combination of capillary pumped heat rejection loops connected to the condenser tube of the secondary loop in the ARCPS. The pumper loops are those designed by the design program. The program calculates the steady state thermodynamic and hydrodynamic balance existing in each heat rejection loop, using as boundary conditions the state of the condensing potassium of the power conversion on the hot side and space temperature.

The calculations begin with the state of the condensing potassium stream as it leaves the turboalternator. As heat is removed by each heat rejection loop, the program calculates the change in the potassium state for calculations to begin on the following pumper loop.

The program calculation sequence is shown in the following block diagram:



FLOW DIAGRAM OF CAPILLARY PUMPER RADIATOR PERFORMANCE PROGRAM

2. INPUT DATA LIST

The following table summarizes the data cards and defines the terms used in the input data block.

LIST OF SYMBOLS USED IN DATA BLOCK

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 1 (12A6)			
ANOTE	Any comment may be typed in Col. 1-72. This comment will be printed on each run.		
CARD 2 (8E9.4)			
HC	Condensation coefficient	Btu/hr-ft^2-°F	--
DX	Thickness of metal between power conversion loop and capillary pump.	ft	--
XKS	Thermal conductivity of metal between power conversion loop and capillary pump.	Btu/hr-ft-°F	--
TA	Temperature of potassium in power conversion loop at start of capillary pump heat rejection area.	°F	
CARD 3 (E9.3,7F9.4)			
QTOT	The initial guess of total heat to be rejected by the capillary pumped radiators.	Btu/hr	--
A	Area of capillary pump liquid to vapor interface (screens) plus area of distribution manifolds (area is normal to heat flow).	ft^2	
TV	Initial guess of capillary pump temperature at liquid to vapor interface.	°F	

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
DXF	Thickness of liquid metal in capillary pump measured from metal to liquid vapor interface.	ft	--
CARD 4 (E9.4,19,6E9.4)			
W	Capillary pumper flow rate	lb/hr	--
I	Trigger to signal choice of capillary pump working fluid (1=Na, 2=k, 3=Ce).	--	--
D1	Capillary pump vapor tube ID	ft	--
D2	Capillary pump liquid tube ID	ft	--
CARD 5 (E9.3,7F9.4)			
R	Capillary screen effective radius.	ft	--
GEE	Acceleration due to gravity	ft/sec^2	
CARD 6 (8E9.4)			
XLIQ	Thickness of capillary pump to radiator liquid tube wall	ft	--
XVAP	Thickness of capillary pump to radiator vapor tube wall	ft	--

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
XPIPL	Length of capillary pump to radiator liquid tube	ft	--
XPIPV	Length of capillary pump to radiator vapor tube	ft.	--
DENM	Density of stainless steel	lb/ft^3	
<u>CARD 7 (8E9.4)</u>			
QUAL	Quality of potassium in power conversion loop at start of capillary pump heat rejection area.		
XLD	Length of the long side of the rectangular power conversion loop condenser channel.	ft	--
WK	Potassium flow rate in power conversion loop.	lb/hr	--
<u>CARD 8 (E9.4,I9,6E9.4)</u>			
PER	Percentage of screened area to total area in capillary pump		
NUM	Number of elements in capillary pumper		
AD	Radiator area convergence criteria		
<u>CARD 9 (8E9.4)</u>			
CLENG	Length of radiator tube to condense all of the vapor.	ft	--
TLTH	Radiator tube length	ft	--

LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 10 (3E9.4,4I5)			
TD	Convergence criteria for iterations on temperature.	°F	--
WD	Convergence criteria for iterations on flow rate.	lb/hr	--
QD	Convergence criteria for iterations on heat flow.	Btu/hr	--
NCC	Number of iterations (upper limit) before program aborts.		
III	Program control trigger. If ($III \leq 0$), program will print out intermediate values of selective variables. If ($III \geq 1$), program will not print these variables.		
NLOOP	Number of capillary pump radiator loops.	--	50
CARD 11 (50I1)			
IRADOP	Program control trigger. If IRADOP = 0, the radiator is inoperative. If IRADOP = 1, the radiator is functioning.		
CARD 12 (8E9.4)	The number of cards depends upon NLOOP. The number will equal the first integer >(NLOOP/8) .		
DPOT	Length of the short side of the rectangular power conversion condenser channel.	ft	--

3. OUTPUT DATA LIST

The principal output data is self-explanitory since the names of all parameters are printed out in full. This print-out is divided into three general groups; the system geometrical parameters, the matched pump loop, thermodynamic and hydrodynamic parameters, and the state conditions within the power conversion loop. This output is given for each heat rejection loop. Table 1 shows the format for this output.

Table 1

Capillary Pump - Program 2

Configuration

Fluid	=		Number of Loops	=	
Temperature at Heat Source	=	°F	Heat Transfer Area	=	FT
Gap Width	=	FT	Capillary Eff. Radius	=	FT
Tubing (Liq)			Tubing (Vapor)		
ID	=	FT	ID	=	FT
Thickness	=	FT	Thickness	=	FT
Length	=	FT	Length	=	FT

Matched System Parameters

N	Flow Rate	Q	Station	Temp.	Press.	Density
2	LB/HR	BTU/HR				
			0			
			1			
			2			
			3			

Cosine Theta =

Radiator - Condensing Tube Length = ft. Total Length = ft.

Power Conversion Loop

Cap-Pump Loop	=	
Condenser		
Diameter	=	FT
Flow Rate	=	LB/HR

Condenser (Cap-P Loop Inlet)			Condenser (Cap-P Loop Exit)		
Temperature	=	°F	Temperature	=	°F
Pressure	=	PSI	Pressure	=	PSI
Velocity	=	FT/HR	Velocity	=	FT/HR
Quality	=		Quality	=	

The program also prints out, automatically, the input data block. The numerical values are printed out in the same order as they appear on the input cards, without identifying symbols. Table 2 shows the format for this output.

Table 2

HC	DXS	XKS	TA				
QTOT	A	TV	DXF				
W	I	D1	D2				
R	GEE	XMTOT					
XLIQ	XVAP	XPIPL	XPIPC	DENM			
QUAL	XLD	WK					
PER	NUM	AD					
CLENG	TLTH						
TD	WD	QD	NCC	III	NLOOP		
IRADOP(II),II=1,NLOOP							
DPOT(I),I=1,NLOOP							
RHOL	CONL	CONH	RHF	RHT	CONF	SUFT	ACONE
R	GAMMA	VISV	HFG	CL	EMM	TS	VISL
RHFO	CONFO	THKR	DCONE	CODF			
TUBN	DI	TFIN	TLTH	ANGLE			

The program may also be requested, by setting III = 0, to print out intermediate values of system parameters and fluid properties for diagnostic purposes.

The main program has a number of error statements which are printed out when the program has trouble completing its mission. These statements are as follows:

1. Program limit (JC or IC) has been exceeded.
2. Radiator - blow thru of vapor at match PT.
3. Inlet velocity GT Mach 1 at match PT.
4. Selected range of W insufficient to obtain Match Point.

5. Initial guess of W is too high to obtain Match Point.
6. Program will not converge during the iteration of loop
7. Program will not converge on TV.
8. Program will not converge on TB.
9. Pumper will not pump. COSINE Theta = _______
10. All the vapor has been condensed.
11. Program will not converge.

```
 I   FOR MAIN,MAIN
C
C CAPILLARY PUMP PROGRAM NUMBER 2
C THIS STEADY STATE PROGRAM COMPUTES THE PERFORMANCE
C OF N CAPILLARY PUMPED LOOPS
C
      DIMENSION X10(24),Y10(24),A10(46,4)
      DIMENSION X13(24),Y13(24),A13(46,4)
      DIMENSION A11(46,4),WSYS(34),DLENG(34)
      DIMENSION PVAP(30),DENV(30)
      DIMENSION ANOTE(12)
      DIMENSION Q(150),IRADOP(150)
      DIMENSION  DPOT(150)
      DIMENSION TGUESS(151)
      REAL LENGTH
C
C CURVE 13 IS SATURATED TEMP OF K AS A FUNCTION OF PRESSURE
C RANGE OF CURVE IS T=1000 F TO 1360 F
C
      N13=12
      DO 6 K=1,N13
      Y13(K)=FLOAT(K)*30.+1000.
    6 X13(K)=PV(2,Y13(K))
      CALL CCDIS (X13,Y13,A13,N13,FN)
      IF(III-1)7,8,8
    7 WRITE(6,361)  (X13(K),Y13(K),K=1,12)
      WRITE(6,201)  (A13(K,1),K=1,46)
    8 CONTINUE
C
      N10=10
      DATA (X10(II),Y10(II),II=1,10)/.01,1.35,.02,1.45,.04,1.6,
     X0.08,1.77,.10,1.85,.20,2.2,.4,2.8,.60,3.3,.80,3.7,1.00,4.0/
      CALL CCDIS (X10,Y10,A10,N10,FN)
C
C NUMBER OF CAP-PUMPER LOOPS LIMITED TO 150
C
   14 READ(5,200)ANOTE
      WRITE(6,206)
      READ(5,201)HC,DXS,XKS,TA
      READ(5,202)QTOT,A,TV,DXF
      READ(5,203)W,I,D1,D2
      READ(5,202)R,GEE,XMTOT
      READ(5,201)XLIQ,XVAP,XPIPL,XPIPV,DENM
      READ(5,201)QUAL,XLD,WK
      READ(5,203)PER,NUM,AD
      READ(5,201)CLENG,TLTH
      READ(5,207)TD,WD,QD,NCC,III,NLOOP
      READ(5,204)(IRADOP(II),II=1,NLOOP)
      READ(5,201)(DPOT(N),N=1,NLOOP)
      WRITE(6,206)ANOTE
      WRITE(6,208)HC,DXS,XKS,TA
      WRITE(6,208)QTOT,A,TV,DXF
      WRITE(6,209)W,I,D1,D2
      WRITE(6,208)R,GEE,XMTOT
```

```
      WRITE(6,208)XLIQ,XVAP,XPIPL,XPIPV,DENM
      WRITE(6,208)QUAL,XLD,WK
      WRITE(6,209)PER,NUM,AD
      WRITE(6,208)CLENG,TLTH
      WRITE(6,210)TD,WD,QD,NCC,III,NLOOP
      WRITE(6,211)(IRADOP(II),II=1,NLOOP)
      WRITE(6,212)(DPOT(N),N=1,NLOOP)
C
C RADIATOR DATA IS READ IN HERE.
      CALL RADP(0.,0.,0.,0.,2.,0.,0.,0.,0,0.)
C
   71 A1=0.7854*D1**2.
      A2=0.7854*D2**2.
      SPRA=5.44
      AHT=A
      KC=0
      DO 90 N=1,NLOOP
      AREA=XLD*DPOT(N)
      WP=2.*(XLD+DPOT(N))
      DA=4.*AREA/WP
      IF(IRADOP(N).GT.0) GO TO 4
      QUE=0.
      Q(N)=0.
      GO TO 80
    4 CONTINUE
      IF(KC.GT.0) GO TO 83
      FAC1=0.805
      MM=0
      TSUB=100.
      QUE=QTOT/FLOAT(NLOOP)
      Q1=QUE
   83 CONTINUE
      DO 20 JJ=1,NCC
      IF(KC.GT.0) GO TO 34
      TK=TA-QUE/(HC*A*PER)
      TI=TK-(QUE*DXS)/(XKS*A*PER)
      DO 73 J=1,NCC
      UF=SKF (I,((TI+TV)/2.))/DXF
      TV1=TI-QUE/(A*PER*UF)
      TVD=(TV1-TV)
      IF (ABS (TV1-TV).LT.TD) GO TO 5
      TV=TV1
      IF (III-1)72,73,73
   72 WRITE(6,250)J,UF,TV1,TVD,TV
   73 CONTINUE
      WRITE(6,382)
      GO TO 14
    5 TV=TV1
   34 CONTINUE
   33 L=L+1
      IF(KC.EQ.0) GO TO 10
      IF(L.LT.NCC) GO TO 35
      WRITE(6,384)L,TV,TGUESS(L),QUE,QRAD,W
      WRITE (6,385)
```

```
      GO TO 14
   35 TV=TV-XINCR
      TGUESS(L)=TV
      IF(KC.EQ.2) GO TO 108
   49 CONTINUE
      XKF=SKF(I,TV)
      UF=HC*XKS*XKF/(DXF*HC*XKS+XKF*XKS+DXS*XKF*HC)
      A=QUE/(UF*(TA-TV)*PER)
      ASUBK=AHT-A
      DO 17 J=1,NCC
      U=SKF(I,((TI+TV)/2.))/DXF
      TI1=TV+QUE/(A*PER*U)
      IF(III-1)63,64,64
   63 WRITE(6,393)J,TI1,TI,U,TV
   64 CONTINUE
      IF(ABS(TI1-TI).LT.TD) GO TO 18
      TI=TI1
   17 CONTINUE
      WRITE(6,390)
      GO TO 14
   18 CONTINUE
      TI=TI1
  108 CONTINUE
      IF(KC.EQ.2)ASUBK=AHT
      DO 19 J=1,NCC
      CP=CPL(2,((TA+TB)/2.))
      XKLM=SKF(2,((TA+TB)/2.))
      HCONV=XKLM*7.20/DA
      XXK=SKF(I,((TA+TB+2.*TV)/4.))
      U=1./((1./HCONV)+(DXS/XKS)+(DXF/XXK))
      TB1=(2.*WK*CP*TA+2.*U*ASUBK*PER*TV-U*ASUBK*PER*TA)/
     X(U*ASUBK*PER+2.*WK*CP)
      IF(ABS(TB1-TB).LT.TD) GO TO 9
      TB=TB1
   19 CONTINUE
      WRITE(6,391)
      GO TO 14
    9 CONTINUE
      TB=TB1
      QSEN=WK*CPL(2,((TA+TB)/2.))*(TA-TB)
      TISC=(QSEN*DXF/(XXK*ASUBK))+TV
      IF(III-1)86,87,87
   86 WRITE(6,389)TV,A,XKF,QUE,TA
      WRITE(6,402)QSEN,TB,WK,ASUBK,U,CP
      WRITE(6,403)XKLM,HCONV
   87 CONTINUE
   10 CONTINUE
C
C ITERATION TO FIND W MATCH
C
      CP=CPL(I,(TV-(TSUB/2.)))
      HHVX=HVX(I,TV)
      PVAP2=PV(I,TV)
      HVAP2=HSAT(I,TV)+HVX(I,TV)
```

```
      HL2=HSAT(I,TV)
      DENV2=RHOV(I.TV)
      SIG2=SIG(I,TV)
      IF (III-1) 74,75,75
   74 WRITE(6,251)PVAP2,TV,HVAP2,HL2,DENV2,SIG2
   75 CONTINUE
      IC=0
      JC=0
      JCC=0
      M=0
   98 CONTINUE
      FAC=FAC1
   50 FAC=FAC-.00491667
      GO TO 52
   51 FAC=FAC+0.005
   52 CONTINUE
      IC=IC+1
      IF(IC.GT.NCC)GO TO 91
   57 W=FAC*Q1/(HHVX+CP*TSUB)
   53 CONTINUE
      IF (III-1) 36,37,37
   36 WRITE(6,365) CP,W,FAC
      WRITE(6,366) IC,M,JC
   37 CONTINUE
      NN=10
      PVAP(1)=PVAP2
      DENV(1)=DENV2
      XMUV2=XMUV (I,TV)
      XVAPP=XVAP/FLOAT (NN)
      DO 76 J=1,NN
      DPVAP=0.263E-11*(XMUV2**0.25)*((W/A2)**1.75)*XVAPP/(DENV(J)*D2
     1**1.25)
      PVAP(J+1)=PVAP(J)-DPVAP
      DENV(J+1)=RHO (I,TV,PVAP(J+1))
   76 CONTINUE
      PT3=PVAP(NN+1)
      DENV3=DENV(NN+1)
      TIN=TV+460.
      WRAD=W/60.
      CALL RADP(TIN,PT3,WRAD,1.,1.,III,T,PTOUT,CONDL,ERR)
      TL1=T-460.
      IF (III-1) 38,39,39
   38 WRITE(6,367) T,PTOUT,PT3,ERR
      WRITE(6,394)TV,CONDL,CLENG,TLTH,WRAD
   39 CONTINUE
      IF(T.LT.0.) GO TO 51
      IF(ERR-1.)54,55,55
   55 JC=JC+1
      IF(JC.GT.9.AND.M.GT.1) GO TO 92
      IF(JC.LT.10) GO TO 50
      IF(JCC.EQ.10) GO TO 92
      FAC1=FAC1-.1
      JCC=JCC+1
      GO TO 98
```

```
54 CONTINUE
   M=M+1
   WSYS(M)=W
   DLENG(M)=CONDL-CLENG
   IF(DLENG(M).GT.0.) GO TO 56
     IF (M.EQ.31) GO TO 95
   FAC=FAC+0.03
   JC=0
   GO TO 57
95 WRITE(6,373)M,WSYS(M),FAC,DLENG(M)
   GO TO 14
56 CONTINUE
   IF(III-1) 28,29,29
28 WRITE(6,368)
   DO 31 MC=1,M
31 WRITE(6,369)MC,DLENG(MC),CLENG,WSYS(MC)
29 CONTINUE
   IF(M-3)43,44,44
43 WRITE(6,374)M,FAC
   IF(MM.LT.5) GO TO 85
   WRITE(6,407) MM,FAC,W,DLENG(M)
   GO TO 14
85 CONTINUE
   MM=MM+1
   FAC1=FAC1-.1
   GO TO 49
44 CONTINUE
   CALL CCDIS(DLENG,WSYS,A11,M,FN)
   CALL CDIS(A11,DLENG,WSYS,0.,W,M,FN)
   DO 58 J=1,NN
   DPVAP=0.263E-11*(XMUV2**0.25)*((W/A2)**1.75)*XVAPP/(DENV(J)*D2
  1**1.25)
   PVAP(J+1)=PVAP(J)-DPVAP
   DENV(J+1)=RHO(I,TV,PVAP(J+1))
58 CONTINUE
   PT3=PVAP(NN+1)
   WRAD=W/60.
   CALL RADP(TIN,PT3,WRAD,1.,1.,III,T,PTOUT,CONDL,ERR)
   IF (ERR.EQ.1) GO TO 93
   IF (ERR.EQ.2) GO TO 94
   TOUT=T-460.
   QRAD=W*(HVX(I,TV)+CPL(I,((TV+TOUT)/2.))*(TV-TOUT))
   IF (III-1) 15,16,16
15 WRITE(6,370) PT3,W,TOUT,QRAD,QUE,PTOUT
   WRITE(6,395)TV,CONDL,CLENG,TLTH
16 CONTINUE
   IF(KC.EQ.0) GO TO 84
   QSUM=QUE+QSEN
   IF(KC.EQ.2)QSUM=QSEN
   IF(III-1)11,12,12
11 WRITE(6,384)L,TV,TGUESS(L),QUE,QRAD,W
   WRITE(6,404)QSUM,QSEN
12 CONTINUE
   IF(ABS(QSUM-QRAD).LT.QD) GO TO 60
```

```
      IF((QSUM-QRAD).GT.0.) GO TO 32
      GO TO 20
   32 XINCR=XINCR/2.
      TV=TGUESS(L-1)
      GO TO 35
   84 CONTINUE
      IF (ABS (QUE-QRAD).LT.QD)GO TO 25
      QUE=QRAD
   20 CONTINUE
      WRITE(6,376) JJ,NCC,N
      GO TO 14
   60 CONTINUE
   25 CONTINUE
      XMUL1=XMUL(I,TOUT)
      RHOL1=RHOL(I,TOUT)
      DPLIQ=0.263E-11*(XMUL1**0.25)*((W/A1)**1.75)*XLIQ/(RHOL1*D1**1.25)
      PT1=PTOUT-DPLIQ
      PTO=PTOUT
      PL1=PT1-(.5*W**2./(RHOL1*A1**2.*602.E+08))
      TSUB=TV-TOUT
      IF(KC.EQ.0) GO TO 59
      IF(KC.EQ.1)TAVG=(TI*A+TISC*ASUBK)/AHT
      IF(KC.EQ.2)TAVG=TISC
      XFRIC=FRIC(AHT,DXF,TAVG,TV,TSUB,PER,NUM,I)
      GO TO 69
   59 CONTINUE
      XFRIC=FRIC(A,DXF,TI,TV,TSUB,PER,NUM,I)
   69 CONTINUE
      CAP=PVAP2-PL1+((((1./(DENV2*A2**2.))-(1./(RHOL1*A1**2)))*(W**2)
     X/4.18E+08)/144.+XFRIC
      COSTH=CAP*R*72./SIG2
      IF(III-1)65,66,66
   65 WRITE(6,396)XFRIC,CAP,XMUL1,RHOL1,DPLIQ,PT1
      WRITE(6,397)PTOUT,PL1,TSUB,COSTH,R,SIG2
   66 CONTINUE
      IF(COSTH.GT.1.) GO TO 22
   80 CONTINUE
C
C  POWER CONVERSION LOOP PIPE LOSES ARE COMPUTED HERE
C
      LENGTH=AHT/(2.*XLD)
      IF(KC.EQ.2) GO TO 101
      VEL=(WK/AREA)*((1-QUAL)/RHOL(2,TA)+QUAL/RHOV(2,TA))
      VELV=1.0*VEL
      TPRESA=SPRA+ ((WK/AREA)**2*(((1.-QUAL)/RHOL(2,TA))+(QUAL/
     XRHOV(2,TA))))/(4.18E+08*144.)
      THETA=SIG(2,TA)*4.18E+08/(XMUV(2,TA)*VEL)
      IF(THETA.LE.57.23) GO TO 40
      IF(THETA.GT.57.23.AND.THETA.LT.327.5) GO TO 41
      CE=0.02
      GO TO 42
   40 CE=0.1
      GO TO 42
   41 CE=4.182/THETA**0.9225
```

```
42 CONTINUE
   FLOWL=(1.-QUAL)*WK
   FLOWV=(QUAL)*WK
   DPDLV=3.1*CE*(XMUV(2,TA)/3600.)**0.2*(FLOWV/3600.)**1.8/
  X(DA**4.8*(RHOV(2,TA))*32.2)
   XTTSQ=((FLOWL/FLOWV)**1.8)*(RHOV(2,TA)/RHOL(2,TA))*
  1((XMUL(2,TA)/XMUV(2,TA))**0.2)
   XTT=SQRT(XTTSQ)
   CALL CDIS(A10,X10,Y10,XTT,PHIG,N10,FN)
   DPDLF=-DPDLV*PHIG**2.
   SVG=1./(RHOV(2,TA))
   SVL=1./(RHOL(2,TA))
   G=(WK/AREA)
   TB =TA
   DO 46 J=1,NCC
   QUALB=QUAL-(QUE/(HVX(2,TA) *WK))+(CPL(2,TA)*(TA-TB))/(HVX(2,TA))
   DPA=(WK/AREA)**2.*(((1.-QUALB)*(1./RHOL(2,TB))+QUALB*(1./RHOV(2,TB
  X)))-((1.-QUAL)*(1./RHOL(2,TA))+QUAL*(1./RHOV(2,TA))))/4.18E+08
   DP=(DPDLF*LENGTH)-DPA
   TPRESB=TPRESA+(DP/144.)
   IF(III-1) 61,62,62
61 WRITE(6,358)XTTSQ,DP,PHIG,DPDLF,DPDLV
   WRITE(6,359)TPRESA,TPRESB,THETA,VEL,CE
62 CONTINUE
   VELB=(WK/AREA)*((1-QUALB)/RHOL(2,TB)+QUALB/RHOV(2,TB))
   VELBV=1.0*VELB
   SPRB=TPRESB-((WK/AREA)**2*(((1.-QUALB)/RHOL(2,TB))+(QUALB/
  XRHOV(2,TB))))/(4.18E+08*144.)
   CALL CDIS(A13,X13,Y13,SPRB,TB1,N13,FN)
   IF (ABS(TB1-TB).LT.0.1) GO TO 47
   IF(III-1) 48,45,45
48 WRITE (6,9916) J,TB1,TB,VELB,SPRB
45 RLG=1.0
   IF(ABS(TB1-TB).GT.20.)RLG=0.3
   TB=(1.-RLG)*TB+RLG*TB1
46 CONTINUE
   WRITE (6,9917)J,TB,TB1
   GO TO 14
47 TB=TB1
   IF(KC.EQ.0) GO TO 68
   QUALB=0.
   TBPR=TB
   DO 26 J=1,NCC
   CP=CPL(2,((TBPR+TB)/2.))
   XKLM=SKF(2.((TBPR+TB)/2.))
   HCONV=XKLM*7.20/DA
   XXK=SKF(I,((TBPR+TB+2.*TV)/4.))
   U=1./((1./HCONV)+(DXS/XKS)+(DXF/XXK))
   ACONV=AHT-A
   TB1=(2.*U*ACONV*PER*TV+2.*WK*CP*TBPR-U*ACONV*PER*TBPR)/
  X(U*ACONV*PER+2.*WK*CP)
   IF(III-1)77,78,78
77 WRITE(6,398)CP,XKLM,HCONV,U,A,ACONV
   WRITE(6,399)TB1,TBPR,TV,WK
```

```
 78 CONTINUE
    IF(ABS(TB1-TB).LT.TD) GO TO 27
    TB=TB1
 26 CONTINUE
    WRITE(6,391)
    GO TO 14
 27 CONTINUE
    TB=TB1
    QSEN=WK*CPL(2,((TB+TBPR)/2.))*(TBPR-TB)
    Q(N)=QUE+QSEN
    VELB=(WK/AREA)*((1-QUALB)/RHOL(2,TB)+QUALB/RHOV(2,TB))
    VELBV=1.0*VELB
    XXMU=XMUL(2,((TA+TB)/2.))
    RHOK=RHOL(2 ,((TA+TB)/2.))
    LENGTH=ACONV/(2.*XLD)
    DPKLIQ=0.263E-11*(XXMU**0.25)*(WK/AREA)**1.75*LENGTH
   X  /(RHOK*DA**1.25)
    TPRESB=TPRESB-DPKLIQ
    SPRB=TPRESB-((WK/AREA)**2)*((1-QUALB)/RHOL(2,TB)+QUALB/RHOV(2,TB))
   X/(4.18E8*144)
    IF(III-1)88,89,89
 88 WRITE(6,390)QSEN,WK,TA,QUE,TB1,TBPR
 89 CONTINUE
    QUE=QUE+QSEN
    KC=2
    L=1
    XINCR=40.
    TGUESS(1)=TV
    GO TO 96
101 CONTINUE
    QUALB=0.
    TB=TA-20.
    DO 102 J=1,NCC
    CP=CPL(2,((TA+TB)/2.))
    XKLM=SKF(2,((TA+TB)/2.))
    HCONV=XKLM*7.20/DA
    XXK=SKF(I,((TA+TB+2.*TV)/4.))
    U=1./((1./HCONV)+(DXS/XKS)+(DXF/XXK))
    TB1=(2.*U*AHT*PER*TV+2.*WK
   1     *CP*TA-U*AHT*PER*TA)/
   2     (U*AHT*PER+2.*WK*CP)
    IF(III-1)103,104,104
103 WRITE(6,398)CP,XKLM,HCONV,U,A,AHT
    WRITE(6,405)TB1,TA,TV,WK
104 CONTINUE
    IF(ABS(TB1-TB).LT.TD) GO TO 105
    TB=TB1
102 CONTINUE
    WRITE(6,391)
    GO TO 14
105 CONTINUE
    TB=TB1
    QUE=WK*CP*(TA-TB)
    Q(N)=QUE
```

```
      VELB=(WK/AREA)*((1-QUALB)/RHOL(2,TB)+QUALB/RHOV(2,TB))
      VELBV=1.0*VELB
      IF(III-1)106,107,107
  106 WRITE(6,406)QUE,WK,TA,TB,CP
  107 CONTINUE
      TPRESA=SPRA+((WK/AREA)**2)*((1-QUAL)/RHOL(2,TA)+QUAL/RHOV(2,TA))/
     X(4.18E8*144)
      XXMU=XMUL(2,((TA+TB)/2.))
      RHOK=RHOL(2,((TA+TB)/2.))
      DPKLIQ=0.263E-11*(XXMU**0.25)*(WK/AREA)**1.75*LENGTH
     X  /(RHOK*DA**1.25)
      TPRESB=TPRESA-DPKLIQ
      SPRB=TPRESB-((WK/AREA)**2)*((1-QUALB)/RHOL(2,TB)+QUALB/RHOV(2,TB))
     X/(4.18E8*144)
      GO TO 96
   68 CONTINUE
      Q(N)=QUE
      QSEN=CPL(2,((TA+TB)/2.))*(TA-TB)*WK
      QCOND=QUE-QSEN
      QUALB=QUAL-(QCOND/(HVX(2,((TA+TB)/2.))*WK))
      IF(QUALB.LT.0.001.AND.QUALB.GT.0.) GO TO 110
      IF(QUALB.GT.0.) GO TO 96
      KC=1
      IF(III.EQ.2)III=0
      L=1
      XINCR=40.
      TGUESS(1)=TV
      QUE=QUAL*WK*HVX(2,TA)
      IF(III-1)23,24,24
   23 WRITE(6,400)
      WRITE(6,401)QUAL,QUALB
   24 CONTINUE
      GO TO 83
  110 CONTINUE
      KC=2
      L=1
      XINCR=40.
      TGUESS(1)=TV
      QUALB=0.
   96 CONTINUE
      WRITE(6,240)
      WRITE(6,241)I,NLOOP,TA,AHT,DXF,R
      WRITE(6,242)D1,D2,XPIPL,XPIPV,XLIQ,XVAP
      WRITE(6,243)
      WRITE(6,9922)
      WRITE(6,352) N,W,QUE,TL1,PT0,RHOL1
      WRITE(6,353)TL1,PT1,RHOL1
      WRITE(6,354)TV,PVAP2,DENV2
      WRITE(6,355)TV,PT3,DENV3
      WRITE(6,9923)COSTH
      WRITE(6,9924)CLENG,TLTH
      WRITE (6,9918)
      WRITE (6,9919) N,DA,WK
      WRITE (6,9920) TA,TB,SPRA,SPRB
```

```
      WRITE (6,9921)VELV,VELBV,QUAL,QUALB
C
C RESET CONDITIONS AT INLET TO NEXT CAPILLARY PUMPED LOOP
C
      SPRA=SPRB
      QUAL=QUALB
      CALL CDIS(A13,X13,Y13,SPRA,TA,N13,FN)
      IF(KC.LT.2) GO TO 90
      L=1
      XINCR=40.
      TGUESS(1)=TV
      TA=TB
      VEL=VELB
      VELV=VELBV
   90 CONTINUE
C
C END POWER CONVERSION LOOP COMPUTATIONS HERE
C
      QTOT = 0.
      DO 81 N=1,NLOOP
   81 QTOT = QTOT+Q(N)
      WRITE(6,356)QTOT
      GO TO 14
   93 WRITE(6,371) TV,W
      GO TO 14
   94 WRITE(6,372) TV,W
      GO TO 14
   91 WRITE(6,362) IC,FAC,W,TSUB,QUE,ERR
      WRITE (6,364)
      GO TO 14
   22 WRITE(6,392)COSTH
      GO TO 14
   92 WRITE(6,363) JC,FAC,W,ERR,TV,PT3
      WRITE (6,364)
      GO TO 14
C
C
  200 FORMAT (12A6)
  201 FORMAT (8E9.4)
  202 FORMAT (E9.3,7F9.4)
  203 FORMAT (E9.4,I9,6E9.4)
  204 FORMAT(50I1)
  206 FORMAT(1H1,12A6)
  207 FORMAT (3E9.4,4I5)
  208 FORMAT(1H ,8E11.4)
  209 FORMAT(1H ,E11.4,I11,2E11.4)
  210 FORMAT(1H ,3E11.4,3I11)
  211 FORMAT(1H ,50I1)
  212 FORMAT(1H ,8E11.4,/,(1X,8E11.4))
  239 FORMAT (I5/(8E9.4))
  240 FORMAT (1H1,39X,28HCAPILLARY PUMPER - PROGRAM 2  ,/)
  241 FORMAT (1H ,6X,14HCONFIGURATION /20X,5HFLUID,24X,1H=,11X,I2,
     1 7X,15HNUMBER OF LOOPS,9X,1H=,11X,I2,/20X,
     230HTEMPERATURE AT HEAT SOURCE   =,F13.5,2H F,5X,
```

```
   318HHEAT TRANSFER AREA,6X,1H=,4X,F9.5,  6H SQ FT,/20X,9HGAP WIDTH,
   X18X,
   42X,1H=,5X,F8.5,3H FT,4X,25HCAPILLARY EFF. RADIUS    =,5X,
   5F8.5,3H FT)
242 FORMAT (1H ,19X,12HTUBING (LIQ),38X,14HTUBING (VAPOR),/
   124X,2HID,23X,1H=,5X,F8.5,3H FT,8X,2HID,18X,1H=,5X,F8.5,3H FT,/
   224X,9HTHICKNESS,16X,1H=,5X,F8.5,3H FT,8X,9HTHICKNESS,11X,1H=,
   35X,F8.5,3H FT,/24X,6HLENGTH,19X,1H=,4X,F9.5,3H FT,8X,6HLENGTH,
   414X,1H=,4XF9.5,3H FT/)
243 FORMAT (1H ,6X,26HMATCHED SYSTEM PARAMETERS )
250 FORMAT(1H ,2HJ=,I2,4E12.4)
251 FORMAT (1H ,4X,5HPVAP=,E10.4,7X,3HTV=,E10.4,4X,6HHVAP2=,E10.4,
   16X,4HHL2=,E10.4,4X,6HDENV2=,E10.4,5X,5HSIG2=,E10.4)
352 FORMAT(1H ,5X,I2,9X,F9.2,6X,E10.4,10X,1H0,9X,F9.2,7X,F7.3,
   17X,E10.4)
353 FORMAT(1H ,51X,1H1,9X,F9.2,7X,F7.3,7X,E10.4)
354 FORMAT(1H ,51X,1H2,9X,F9.2,7X,F7.3,7X,E10.4)
355 FORMAT(1H ,51X,1H3,9X,F9.2,7X,F7.3,7X,E10.4)
356 FORMAT (1H ,6X,24HSYSTEM HEAT REJECTION = ,E10.4,
   110H   BTU/HR )
358 FORMAT (1H ,10H    XTTSQ=,E10.4,10H       DP=,E10.4,
   110H      PHIG=,E10.4,10H    DPDLF=,E10.4,10H    DPDLV=,E10.4)
359 FORMAT (1H ,10H   TPRESA=,E10.4,10H   TPRESB=,E10.4,
   110H     THETA=,E10.4,10H      VEL=,E10.4,10H       CE=,E10.4)
361 FORMAT (1H , 7HX13,Y13  /(6E11.4)/(6E11.4)/(6E11.4))
362 FORMAT (1H ,6X,3HIC=,I10,6X,4HFAC=,E10.4,8X,2HW=,E10.4,
   X5X,5HTSUB=,E10.4,6X,4HQUE=,E10.4,6X,4HERR=,E10.4)
363 FORMAT (1H ,6X,3HJC=,I10,6X,4HFAC=,E10.4,8X,2HW=,E10.4,6X,
   X4HERR=,E10.4,7X,3HTV=,E10.4,6X,4HPT3=,E10.4)
364 FORMAT (1H ,44HA PROGRAM LIMIT (JC OR IC) HAS BEEN EXCEEDED   )
365 FORMAT (1H ,6X,3HCP=,E10.4,8X,2HW=,E10.4,6X,4HFAC=,E10.6     )
366 FORMAT (1H ,6X,3HIC=,I10,8X,2HM=,I10,7X,3HJC=,I10)
367 FORMAT (1H ,7X,2HT=,E10.4,4X,6HPTOUT=,E10.4,6X,4HPT3=,E10.4,
   X6X,4HERR=,E10.4)
368 FORMAT(1H ,8X,1HM,9X,5HDLENG,10X,5HCLENG,10X,4HWSYS    )
369 FORMAT(1H ,I10,5X,E10.4,5X,E10.4,5X,E10.4      )
370 FORMAT (1H ,5X,4HPT3=,E10.4,8X,2HW=,E10.4,5X,5HTOUT=,E10.4,
   X5X,5HQRAD=,E10.4,6X,4HQUE=,E10.4,4X,6HPTOUT=,E10.4)
371 FORMAT (1H ,42H RADIATOR - BLOW THRU OF VAPOR AT MATCH PT ,
   X7X,3HTV=,E10.4,8X,2HW=,E10.4 )
372 FORMAT (1H ,36HINLET VELOCITY GT MACH 1 AT MATCH PT   ,
   X7X,3HTV=,E10.4,8X,2HW=,E10.4 )
373 FORMAT(1H ,7X,2HM=,I10,2X,8HWSYS(M)=,E10.4,6X,4HFAC=,E10.4,
   X1X,9HDLENG(M)=,E10.4,/,
   X56H SELECTED RANGE OF W INSUFFICIENT TO OBTAIN MATCH POINT.    )
374 FORMAT (1H ,2HM=,I5,41HINITIAL GUESS OF W IS TOO HIGH TO OBTAIN ,
   X18HMATCH POINT. FAC= ,F4.2      )
376 FORMAT(1H ,6X,3HJJ=,I10,6X,4HNCC=,I10,5X,
   X54HPROGRAM WILL NOT CONVERGE DURING THE ITERATION OF LOOP ,I5 )
382 FORMAT(1H ,32HPROGRAM WILL NOT CONVERGE ON TV.           )
384 FORMAT(1H ,7X,2HL=,I10,7X,3HTV=,E10.6,5X,5HT(L)=,E10.6,
   X6X,4HQUE=,E10.4,5X,5HQRAD=,E10.4,8X,2HW=,E10.4   )
385 FORMAT (1H ,35HPROGRAM WILL NOT CONVERGE ON TV                  )
389 FORMAT(1H ,6X,3HTV=,E10.6,8X,2HA=,E10.4,6X,4HXKF=,E10.4,
```

```
     X6X,4HQUF=,F10.4,7X,3HTA=,E10.4     )
 390 FORMAT(1H ,4X,5HQSEN=,E10.6,7X,3HWK=,E10.4,7X,3HTA=,E10.4,
     X6X,4HQUF=,E10.6,6X,4HTB1=,E10.6,5X,5HTBPR=,E10.6    )
 391 FORMAT(1H ,32HPROGRAM WILL NOT CONVERGE ON TB.        )
 392 FORMAT(1H ,35HPUMPER WILL NOT PUMP. COSINE THETA= ,F6.3    )
 393 FORMAT(1H ,7X,2HJ=,I10,6X,4HTI1=,E10.6,7X,3HTI=,E10.6,
     X8X,2HU=,E10.4,7X,3HTV=,E10.6    )
 394 FORMAT(1H ,6X,3HTV=,E10.6,4X,6HCONDL=,E10.4,4X,6HCLENG=,E10.4,
     X5X,5HTLTH=,E10.4,5X,5HWRAD=,E10.4    )
 395 FORMAT(1H ,6X,3HTV=,E10.6,4X,6HCONDL=,E10.4,4X,6HCLENG=,E10.4,
     X5X,5HTLTH=,F10.4    )
 396 FORMAT(1H ,3X,6HXFRIC=,E10.4,6X,4HCAP=,E10.4,4X,6HXMUL1=,E10.4,
     X4X,6HRHOL1=,E10.4,4X,6HDPLIQ=,E10.4,6X,4HPT1=,E10.4    )
 397 FORMAT(1H ,4X,6HPTOUT=,E10.4,6X,4HPL1=,E10.4,5X,5HTSUB=,E10.4,
     X4X,6HCOSTH=,F10.4,8X,2HR=,E10.4,5X,5HSIG2=,E10.4    )
 398 FORMAT(1H ,6X,3HCP=,E10.4,5X,5HXKLM=,E10.4,4X,6HCONV=,E10.4,
     X8X,2HU=,E10.4,8X,2HA=,E10.4,4X,6HACONV=,E10.4    )
 399 FORMAT(1H ,5X,4HTB1=,E10.6,5X,5HTBPR=,E10.6,7X,3HTV=,E10.6,
     X7X,3HWK=,F10.4     )
 400 FORMAT(1H0,33HALL THE VAPOR HAS BEEN CONDENSED.  ,/)
 401 FORMAT(1H ,4X,5HQUAL=,E10.4,4X,6HQUALB=,E10.4    )
 402 FORMAT(1H ,4X,5HQSEN=,E10.6,7X,3HTB=,E10.6,7X,3HWK=,E10.4,
     X4X,6HASUBK=,E10.4,8X,2HU=,E10.6,7X,3HCP=,E10.4     )
 403 FORMAT(1H ,4X,5HXKLM=,E10.4,4X,6HHCONV=,E10.4     )
 404 FORMAT(1H ,4X,5HQSUM=,E10.6,5X,5HQSEN=,E10.6    )
 405 FORMAT(1H ,5X,4HTB1=,E10.6,7X,3HTA=,E10.6,7X,3HTV=,E10.6,
     X7X,3HWK=,E10.4    )
 406 FORMAT(1H ,5X,4HQUE=,E10.4,7X,3HWK=,E10.4,7X,3HTA=,E10.6,
     X7X,3HTB=,E10.6,7X,3HCP=,E10.4    )
 407 FORMAT(1H ,3HMM=,I3,2X,4HFAC=,F6.3,2X,2HW=,E10.6,2X,9HDLENG(M)=,
     XE10.4,3X,27H PROGRAM WILL NOT CONVERGE.    )
9916 FORMAT (1H ,4HI = ,I5, 8H    TB1= ,F13.4,6H    TB=,F13.4,
     13H    VFLB=,F13.4,8H    SPRB=,F13.4)
9917 FORMAT (1H ,2HJ=,I5,7H    TB= ,F13.4,8H    TB1= ,F13.4)
9918 FORMAT (1H ,6X,21HPOWER CONVERSION LOOP    )
9919 FORMAT (1H ,19X,15HCAP-PUMPER LOOP ,14X1H=,I6,/20X,9HCONDENSER/,
     124X,8HDIAMETER,17X,1H=,F11.4,3X2HFT/24X,9HFLOW RATE,16X,1H=,
     2F11.4,2X,5HLB/HR    )
9920 FORMAT (1H ,19X,28HCONDENSER (CAP-P LOOP INLET),22X,
     127HCONDENSER (CAP-P LOOP EXIT)/24X,11HTEMPERATURE,14X,1H=,F11.4,
     24X1HF,8X,11HTEMPERATURE,14X,1H=,F11.4,4X,1HF/24X,8HPRESSURE,
     317X,1H=,F11.4,2X,3HPSI,8X,8HPRESSURE,17X,1H=,F11.4,2X,3HPSI   )
9921 FORMAT (1H ,23X,8HVELOCITY,17X,1H=,E11.4,2X,5HFT/HR,6X,
     18HVELOCITY,17X,1H=,E11.4,2X,5HFT/HR,/24X,7HQUALITY,18X,1H=,
     2F11.4,13X,7HQUALITY,18X,1H=,F11.4    )
9922 FORMAT (1H ,6X,1HN,10X,9HFLOW RATE,10X,1HQ,11X,
     18H STATION,10X,5HTEMP.,9X,6HPRESS.,8X,7HDENSITY,/
     220X,5HLB/HR,10X,6HBTU/HR,24X,5HDEG F,11X,
     33HPSI,10X,8HLB/CU-FT        )
9923 FORMAT(1H ,19X,15HCOSINE THETA = ,F5.3       )
9924 FORMAT (1H ,19X,36HRADIATOR - CONDENSING TUBE LENGTH = ,F7.3,
     X2X,21HFT,    TOTAL LENGTH = ,F7.3,2X,3HFT.    )
      END
```

5. AN EXAMPLE CASE

The case used as an example for this program is a six loop heat rejection system using potassium as the working fluid. The condensing potassium channel is 20 x 2.5 in. rectangular cross section.

The input data listing is shown in the following Table. The corresponding computer output is given on the succeeding six pages.

EXAMPLE CASE INPUT

```
RUN  4  SIX LOOPS
    .1050+05   .417 -02   .3150+02   .1220+04
    .5240+07   .1260+02   .1420+04   .1250-02
    .1000+04          2   .9530-01   .2830+00
    .7000-04   .0030      .1000+01
    .2500+02   .5000+01   .2540-02   .2500-02   .5360+03
    .8900+00   .1667+01   .6480+04
    .9000+00         16   .1000-01
    .1750+02   .1831+02
    .1000+00   .1000-01   .1000+00        200          1          6
111111
    .2080+00   .2031+03   .2190+10   .2050+00   .2030+00   .2080+00
    .4730+02   .1990+02   .1050+03   .5370+03   .5000+03   .1790+03   .5000-02  -.1745+00
   -.3950+02   .1610+01   .1230-04   .6720+03   .1000+00   .9000+00   .5650+03   .9500-04
    .5000+03   .1370+03   .4010+00   .7320+01   .1000+01
    .2510+02   .1150+01   .3030-01   .1930+02   .6000+02
END DATA INPUT.
```

EXAMPLE CASE OUTPUT

CAPILLARY PUMPER - PROGRAM 2

```
CONFIGURATION
            FLUID                           =          2      NUMBER OF LOOPS             =          6
            TEMPERATURE AT HEAT SOURCE      =   1220.00000 F  HEAT TRANSFER AREA          =   12.80000 SQ FT
            GAP WIDTH                       =      .00125 FT  CAPILLARY EFF. RADIUS       =     .00007 FT
            TUBING (LIQ)                                      TUBING (VAPOR)
                ID                          =      .09530 FT      ID                      =     .28300 FT
                THICKNESS                   =      .00250 FT      THICKNESS               =     .00250 FT
                LENGTH                      =    25.00000 FT      LENGTH                  =    5.00000 FT

MATCHED SYSTEM PARAMETERS
N         FLOW RATE        Q            STATION      TEMP.       PRESS.      DENSITY
            LB/HR        BTU/HR                      DEG F       PSI         LB/CU-FT
1         1115.08        .9678+06          0        1045.41      4.259       .4505+02
                                           1        1045.41      4.232       .4505+02
                                           2        1195.51      4.535       .1018-01
                                           3        1195.51      4.455       .9801-02
            COSINE THETA =   .818
            RADIATOR -- CONDENSING TUBE LENGTH =  17.500  FT,   TOTAL LENGTH =  18.300  FT.
POWER CONVERSION LOOP
            CAP-PUMPER-LOOP                 =     1
            CONDENSER
                DIAMETER                    =        .3699    FT
                FLOW RATE                   =    6480.0000    LB/HR
            CONDENSER (CAP-P-LOOP INLET)                      CONDENSER (CAP-P LOOP EXIT)
                TEMPERATURE                 =    1220.0000    F       TEMPERATURE          =  1203.5673    F
                PRESSURE                    =       5.4400    PSI     PRESSURE             =     4.7754   PSI
                VELOCITY                    =     .1412+07    FT/HR   VELOCITY             =   .1247+07   FT/HR
                QUALITY                     =        .8900            QUALITY              =      .7132
```

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CAPILLARY PUMPER - PROGRAM 2

CONFIGURATION

FLUID	=	2	NUMBER OF LOOPS	=	6
TEMPERATURE AT HEAT SOURCE	=	1203.56734 F	HEAT TRANSFER AREA	=	12.80000 SQ FT
GAP WIDTH	=	.00125 FT	CAPILLARY EFF. RADIUS	=	.00007 FT
TUBING (LIQ)			TUBING (VAPOR)		
ID	=	.09530 FT	ID	=	.28300 FT
THICKNESS	=	.00250 FT	THICKNESS	=	.00250 FT
LENGTH	=	25.00000 FT	LENGTH	=	5.00000 FT

MATCHED SYSTEM PARAMETERS

N	FLOW RATE LB/HR	Q BTU/HR	STATION	TEMP. DEG F	PRESS. PSI	DENSITY LB/CU-FT
2	1063.95	.9263+06	0	987.66	3.816	.4520+02
			1	987.66	3.791	.4520+02
			2	1180.16	4.104	.9276-02
			3	1180.16	4.023	.8938-02

COSINE THETA = .821

RADIATOR CONDENSING TUBE LENGTH = 17.500 FT, TOTAL LENGTH = 18.300 FT.

POWER CONVERSION LOOP

CAP-PUMPER LOOP	=	2	
CONDENSER			
DIAMETER	=	.3699	FT
FLOW RATE	=	6480.0000	LB/HR

CONDENSER (CAP-P LOOP INLET)				CONDENSER (CAP-P LOOP EXIT)			
TEMPERATURE	=	1203.5673	F	TEMPERATURE	=	1186.5607	F
PRESSURE	=	4.7751	PSI	PRESSURE	=	4.2795	PSI
VELOCITY	=	.1248+07	FT/HR	VELOCITY	=	.1055+07	FT/HR
QUALITY	=	.7132		QUALITY	=	.5448	

CAPILLARY PUMPER - PROGRAM 2

CONFIGURATION

FLUID	= 2		NUMBER OF LOOPS	= 6
TEMPERATURE AT HEAT SOURCE	= 1186.56075 F		HEAT TRANSFER AREA	= 12.80000 SQ FT
GAP WIDTH	= .00125 FT		CAPILLARY EFF. RADIUS	= .00007 FT
TUBING (LIQ)			TUBING (VAPOR)	
ID	= .09530 FT		ID	= .28300 FT
THICKNESS	= .00250 FT		THICKNESS	= .00250 FT
LENGTH	= 25.00000 FT		LENGTH	= 5.00000 FT

MATCHED SYSTEM PARAMETERS

N	FLOW RATE LB/HR	Q BTU/HR	STATION	TEMP. DEG F	PRESS. PSI	DENSITY LB/CU-FT
3	1016.14	.8873+06	0	1017.82	3.387	.4535+02
			1	1017.82	3.363	.4535+02
			2	1164.17	3.692	.8409-02
			3	1164.17	3.610	.8094-02

COSINE THETA = .832

RADIATOR - CONDENSING TUBE LENGTH = 17.500 FT, TOTAL LENGTH = 18.300 FT.

POWER CONVERSION LOOP

CAP-PUMPER LOOP = 3

CONDENSER

DIAMETER	= .3699 FT
FLOW RATE	= 6480.0000 LB/HR

CONDENSER (CAP-P LOOP INLET)		CONDENSER (CAP-P LOOP EXIT)	
TEMPERATURE	= 1186.5607 F	TEMPERATURE	= 1175.0104 F
PRESSURE	= 4.2795 PSI	PRESSURE	= 3.9676 PSI
VELOCITY	= .1056+07 FT/HR	VELOCITY	= .7953+06 FT/HR
QUALITY	= .5448	QUALITY	= .3827

CAPILLARY PUMPER - PROGRAM 2

CONFIGURATION

FLUID	=	2	NUMBER OF LOOPS	=	6
TEMPERATURE AT HEAT SOURCE	=	1175.01044 F	HEAT TRANSFER AREA	=	12.80000 SQ FT
GAP-WIDTH	=	.00125 FT	CAPILLARY EFF. RADIUS	=	.00007 FT
TUBING (LIQ)			TUBING (VAPOR)		
ID	=	.09530 FT	ID	=	.28300 FT
THICKNESS	=	.00250 FT	THICKNESS	=	.00250 FT
LENGTH	=	25.00000 FT	LENGTH	=	5.00000 FT

MATCHED SYSTEM PARAMETERS

N	FLOW RATE LB/HR	Q BTU/HR	STATION	TEMP. DEG F	PRESS. PSI	DENSITY LB/CU-FT
4	986.41	.8628+06	0	998.22	3.111	.4544+02
			1	998.22	3.089	.4544+02
			2	1153.26	3.431	.7658-02
			3	1153.26	3.347	.7556-02

COSINE THETA = .644

RADIATOR---CONDENSING TUBE LENGTH = 17.500 FT, TOTAL LENGTH = 18.300 FT.

POWER CONVERSION LOOP

CAP-PUMPER-LOOP	=	4
CONDENSER		
DIAMETER	=	.3699 FT
FLOW RATE	=	6480.0000 LB/HR

CONDENSER (CAP-P-LOOP-INLET)			CONDENSER (CAP-P LOOP EXIT)		
TEMPERATURE	=	1175.0104 F	TEMPERATURE	=	1171.1236 F
PRESSURE	=	3.9676 PSI	PRESSURE	=	3.8669 PSI
VELOCITY	=	.7959+06 FT/HR	VELOCITY	=	.4760+06 FT/HR
QUALITY	=	.3827	QUALITY	=	.2235

CAPILLARY PUMPER - PROGRAM 2

CONFIGURATION

FLUID	=	2	NUMBER OF LOOPS	=	6
TEMPERATURE AT HEAT SOURCE	=	1171.12364 F	HEAT TRANSFER AREA	=	12.80000 SQ FT
GAP WIDTH	=	.00125 FT	CAPILLARY EFF. RADIUS	=	.00007 FT
TUBING (LIQ)			TUBING (VAPOR)		
ID	=	.09530 FT	ID	=	.28300 FT
THICKNESS	=	.00250 FT	THICKNESS	=	.00250 FT
LENGTH	=	25.00000 FT	LENGTH	=	5.00000 FT

MATCHED SYSTEM PARAMETERS

N	FLOW RATE LB/HR	Q BTU/HR	STATION	TEMP. DEG F	PRESS. PSI	DENSITY LB/CU-FT
5	976.89	.8550+06	0	1027.92	3.022	.4548+02
			1	1027.92	3.000	.4548+02
			2	1149.58	3.346	.7679-02
			3	1149.58	3.262	.7377-02

COSINE THETA = .848

RADIATOR CONDENSING TUBE LENGTH = 17.500 FT, TOTAL LENGTH = 18.300 FT.

POWER CONVERSION LOOP

CAP-PUMPER LOOP	=	5					
CONDENSER							
DIAMETER	=	.3699	FT				
FLOW RATE	=	6480.0000	LB/HR				
CONDENSER (CAP-P LOOP INLET)				CONDENSER (CAP-P LOOP EXIT)			
TEMPERATURE	=	1171.1236	F	TEMPERATURE	=	1175.8753	F
PRESSURE	=	3.8669	PSI	PRESSURE	=	3.9903	PSI
VELOCITY	=	.4762+06	FT/HR	VELOCITY	=	.1322+06	FT/HR
QUALITY	=	.2235		QUALITY	=	.0637	

M= 1INITIAL GUESS OF W IS TOO HIGH TO OBTAIN MATCH POINT. FAC= .80

M= 1INITIAL GUESS OF W IS TOO HIGH TO OBTAIN MATCH POINT. FAC= .70

M= 1INITIAL GUESS OF W IS TOO HIGH TO OBTAIN MATCH POINT. FAC= .60

CAPILLARY PUMPER - PROGRAM 2

```
CONFIGURATION
          FLUID                       =          2        NUMBER OF LOOPS          =        6
          TEMPERATURE AT HEAT SOURCE  =  1175.87526 F     HEAT TRANSFER AREA       =   12.80000 SQ FT
          GAP WIDTH                   =     .00125 FT     CAPILLARY EFF. RADIUS    =     .00007 FT
          TUBING (LIQ)                                    TUBING (VAPOR)
             ID                       =     .09530 FT        ID                    =     .28300 FT
             THICKNESS                =     .00250 FT        THICKNESS             =     .00250 FT
             LENGTH                   =   25.00000 FT        LENGTH                =    5.00000 FT

MATCHED SYSTEM PARAMETERS
N        FLOW RATE        Q           STATION       TEMP.       PRESS.      DENSITY
         LB/HR          BTU/HR                      DEG F       PSI         LB/CU-FT
6        650.32        .5872+06         0           884.13      .847        .4669+02
                                        1           884.13      .837        .4669+02
                                        2           1021.08     1.296       .3113-02
                                        3           1021.08     1.197       .2933-02
          COSINE THETA =   .941
          RADIATOR    CONDENSING TUBE LENGTH =  17.500  FT,   TOTAL LENGTH =  18.300  FT.
POWER CONVERSION LOOP
          CAP-PUMPER LOOP            =      6
          CONDENSER
             DIAMETER                =       .3699    FT
             FLOW RATE               =   6480.0000    LB/HR
          CONDENSER (CAP-P LOOP INLET) --                 CONDENSER (CAP-P LOOP EXIT)
             TEMPERATURE             =   1175.8753    F      TEMPERATURE           =   984.5262   F
             PRESSURE                =     -3.9903    PSI    PRESSURE              =     4.0671   PSI
             VELOCITY                =    .1321+06    FT/HR  VELOCITY              =   .4142+03   FT/HR
             QUALITY                 =     -.0637            QUALITY               =      .0000
SYSTEM HEAT REJECTION =   .5087+07   BTU/HR
```

APPENDIX E

RADIATOR OPTIMIZATION PROGRAM

The radiator optimization program is described in this appendix. Section 1, Program Abstract, gives a statement of the program's purpose and the calculation sequence. Section 2, Input Data List, defines the items required in the input data block. Section 3, Output Data List, defines terms used in the output data block. Section 4, Program Listing, gives a print-out of the program card deck. Section 5, An Example Case, gives the input and output data for an example case.

1. PROGRAM ABSTRACT

The program designs a number of radiators for a range of internal parameters input by the user and does not internally optimize the radiator weight. Optimum radiators are then determined external to the program by the user.

The program is integrated into the SINDA network analysis program, reference 10, using the General option with a "Load and Go Subroutine." Input is through the SINDA Array Data block and five data cards are read by the subroutine using standard FORTRAN Read Statements.

2. INPUT DATA LIST

The following table summarizes the data cards and defines the terms used in the input data block.

Table E-1. LIST OF SYMBOLS USED IN DATA BLOCK

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 1 (8E10.5)			
RHOL	Liquid density.	lb/ft^3	
CONL	Liquid thermal conductivity.	Btu/hr-ft-°F	
CONH	Condensing heat transfer coefficient	$Btu/hr\text{-}ft^2\text{-}°F$	
DP	Total static pressure drop	psia	
PI	Inlet pressure	psia	
TI	Inlet temperature	°R	
RHF	Density inner fin material	lb/ft^3	
RHT	Density tube material	lb/ft^3	
CARD 2 (7E10.5)			
CONF	Thermal conductivity inner fin material	Btu/hr-ft-°F	
SUFT	Working fluid surface tension	lb/ft	
ACONE	Cone half angle	radians	
TOT	Outlet temperature	°R	
WD	Flow rate	lb/minute	

Table E-1. LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOLS	DESCRIPTION	UNITS	LIMIT
R	Gas constant	ft/°F	
GAMMA	Specific heat ratio		
CARD 3 (8E10.5)			
VISV	Vapor viscosity	lb/ft-sec	
HFG	Heat of vaporization	Btu/lb	
CL	Liquid specific heat	Btu/lb-°F	
DCONE	Cone diameter at inlet	ft	
EMM	Fin/tube emittance		
TS	Equivalent sink temperature (565)	°R	
CML	Maximum tube length (23.1')	ft	
CODF	Fluid type code: 1. = potassium, 2. = sodium, 3. = cesium		
CARD 4 (4E10.5)			
VISL	Liquid viscosity	lb/ft-sec	
RRFO	Density of outer fin material	lb/ft^3	
CONFO	Conductivity of outer fin material	Btu/hr-ft-°F	
THKR	Thickness ratio $= \frac{\tau_{outer\ total}}{\tau_{inner}}$		

Table E-1. LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
CARD 5 (6E10.5)			
DI1	First tube inside diameter	inches	
DDI1	Increment on inside diameter	inches	
DN	Number of tube diameter combinations		
TT1	First number of tubes		
DNT	Increment on number of tubes		
TN	Number of tube number combinations		
SINDA TABLE INPUT ARRAYS			
A1	Lockart-Martinelli; correlation		
A2	Tsat v.s. P for potassium	°R, psia	
A3	Tsat v.s. P for sodium	°R, psia	
A4	Tsat v.s. P for cesium	°R, psia	
A5	Fin effectiveness v.s. N_c relation		
A6	N_c v.s. fin effectiveness relation		

3. OUTPUT DATA LIST

The following table defines terms used in the computer output. It also gives the meanings of error statements built into the program.

Table E-2. Output Data Definitions

DI	=	Tube ID at inlet to radiator (inches)
DO	=	Tube ID at outlet of radiator (inches)
NTUBES	=	Total number of tubes
TFIN	=	Fin thickness (inches)
FEFF	=	Fin effectiveness, approximate
QTOTL	=	Total heat rejection (Btu/hr)
QCOND	=	Heat rejected in condenser section (Btu/hr)
QSCLR	=	Heat rejected in subcooler section (Btu/hr)
LTOT	=	Total radiator tube length (ft)
LCOND	=	Condenser section tube length (ft)
LSCLR	=	Subcooler section tube length (ft)
ATOT	=	Total radiator area (ft^2)
FINW	=	Fin weight (lbs)
TINSUB	=	Inlet temperature to subcooler (°R)
DPIH	=	Inlet header pressure drop (psia)
WILH	=	Inlet header weight (lbs)

TUBE WALL Thickness for 0, 1, 2, 3, 4, 5 penetrations (inches)

TUBW0	=	Tube weight - 0 penetrations (lbs)
TUBW1	=	Tube weight - 1 penetrations (lbs)
TUBW2	=	Tube weight - 2 penetrations (lbs)
TUBW3	=	Tube weight - 3 penetrations (lbs)
TUBW4	=	Tube weight - 4 penetrations (lbs)
TUBW5	=	Tube weight - 5 penetrations (lbs)

Table E-2. Output Data Definitions (Cont'd)

W0 = Total radiator weight 0 penetrations (lbs)

W1 = Total radiator weight 1 penetrations (lbs)

W2 = Total radiator weight 2 penetrations (lbs)

W3 = Total radiator weight 3 penetrations (lbs)

W4 = Total radiator weight 4 penetrations (lbs)

W5 = Total radiator weight 5 penetrations (lbs)

ACTUAL COND FIN Q = Calculated heat rejection from fin in condenser section (Btu/hr)

REQUIRED FIN Q = Required fin heat rejection. For valid solution this quantity and the above must be nearly equal.

WLQ = Weight of liquid inventory in Subcooler Section (lbs)

Error Messages

If a valid solution cannot be determined for the combination of tube number and diameter chosen, error messages are printed and the program passes on to the next combination. Meaning of these messages is summarized below.

1.0 Required η greater than 0.99

2.0 Length calculated greater than available.

3.0 Available two phase ΔP less than zero.

4.0 Too many tubes to fit on circumference.

5.0 Q subcooler less than zero.

6.0 Inlet velocity greater than sonic velocity.

7.0 Fin thickness less than .001".

8.0 Fin thickness greater than 0.1".

4. PROGRAM LISTING

A listing of the program card deck is given in the following section.

```
 FOR CONRAD,CONRAD
      SUBROUTINE CONRAD(A1,A2,A3,A4,A5,A6)
      DIMENSION DPOL(10)
      WRITE(6,900)
900   FORMAT(48H1    **** CONDENSING RADIATOR SIZING PROGRAM ****)
230   READ(5,105) RHOL,CONL,CONH,DP,PI,TI,RHF,RH1
      READ(5,110)       CONF,SUFT,ACONE,TOT,WD,R,GAMMA
110   FORMAT(7E10.5)
      READ(5,105) VISV,HFG,CL,DCONE,EMM,TS,CML,CODF
      READ(5,909) VISL,RHFO,CONFO,THKR
909   FORMAT(4E10.5)'
      READ(5,606) DI1,DDI1,DN,TT1,DNT,TM
606   FORMAT(6F10.5)
105   FORMAT(9E10.5)
      WRITE(6,901)
901   FORMAT(72H0     RHOL         HCONDS         KLIQ         HCONDP         RHFI
     C          RHFO     )
      WRITE(6,902) RHOL,CONH,CONL,RHF,RPT,RHFO
902   FORMAT(1H ,1X,F7.1,5F12.1)
      WRITE(6,903)
903   FORMAT(72H        TIN         THKR         KFINI         KFINO         TOUT
     C          FLOW     )
      WRITE(6,902)TI,THKR,CONF,CONFO,TOT,WD
      WRITE(6,904)
904   FORMAT(72H         R         GAMMA         HFG         TSNK         DCON
     CE          LMAX     )
      WRITE(6,902) R,GAMMA,HFG,TS,DCONE,CML
      WRITE(6,905)
905   FORMAT(72H        DP         PI         SURFT         ACONE         VISV
     C          VISL     )
      WRITE(6,906) DP,PI,SUFT,ACONE,VISV,VISL
906   FORMAT(1H ,1X,F7.2,F12.2,F12.4,F12.4,F12.6,F12.6)
      WRITE(6,907)
907   FORMAT(72H        CPL         EMIS         CODF
     C                                     )
      WRITE(6,908) CL,EMM,CODF
908   FORMAT(2H    ,F7.2,F12.3,F12.1)
      TMINM=.03
      RHF=(THKR*RHFO+RHF)/(1.+THKR)
      CONF = (THKR*CONFO+CONF)/(1.+THKR)
      XI=1.0
      DI=DI1
      DO=DI1
      DO=DO
      A10=.01
      A11=.15
      A12=.43
      A13=.83
      A14=1.28
      A15=1.80
      XD=0.1
240   TUBN=TT1
      XT=0.1
      RHOV=PI*144./(R*TI)
```

Figure E-1. Optimization Program Listing

```
      VSONC=5.67*SQRT(GAMMA*R*TI)
220   VIN=WO/(60.*TUBN*RHOV*3.14*DI**2)*4.*144.
      IF(VIN.GT.VSONC) GO TO 800
      DINH=.5*DI*SQRT(TUBN)
      REN=VIN*RHOV*DINH/(12.*VISV)
      DPIH=.316*3.14*DCONF*RHOV*VIN**2/(REN**.25*DINH*24.*32.2)
      DPIH=DPIH/4.
      REVIN=REN*DI/DINH
      DPLC=DP-DPIH+RHOV*VIN*VIN/9260.
      IF(DPLC.LT.0.0) GO TO 10
      X2=DO*TUBN/12./3.14
      IF(X2.GE.DCONF) GO TO 70
      X=1.05
      PT1=0.0
      PT2=0.0
      DO 20 I=1,10
      J1=I
      X=X-.1
      D=DI-(1.-X)*(DI-DO)
      DIN=D
      V=VIN*X*(DI/D)**2
      THETA=SURFT*32.2/(VISV*V)
      RE=V*D/12.*RHOV/VISV
      IF(THETA.GT.57.23.AND.THETA.LT.327.59) CF=4.182/THETA**.9225
      IF(THETA.GE.327.59) CF=.02
      IF(THETA.LE.57.23) CF=.1
      A=(((1.-X)/X)**1.8*RHOV/RHOL*(VISL/VISV)**.2)**.5
      CALL DIDFG1(A,A1 ,B)
      PT1=PT1+.1*CF*2./(RE)**.2*RHOV*V**2/(D*32.2*12.)*B**2
      DPOL(I)=(PT1-PT2)*10.
      PT2=PT1
   20 CONTINUE
      ELC=DPLC/PT1
      IF(ELC.GT.CML) GO TO 30
      IF(CODE.LT.1.5) GO TO 40
      IF(CODE.LT.2.5) GO TO 50
      PO=PI-DP
      CALL DIDFG1(PO,A4,TO)
      GO TO 60
   40 PO=PI-DP
      CALL DIDFG1(PO,A2,TO)
      GO TO 60
   50 PO=PI-DP
      CALL DIDFG1(PO,A3,TO)
   60 CONTINUE
      QCOND=HFG*WO+WO*CL*(TI-TO)
      ELC=12.*ELC
      ATUB=SQRT((DI-DO)**2/4.+ELC**2)*3.14*(DI+DO)/2./144.*TUBN
      ELC=ELC/12.
      DLT=QCOND*60./(ATUB*CONH)
      QCOND=QCOND*60.
      TAVW=(TI+TO)/2.-DLT
      ATUBO=TUBN*(DI+DO)/24.*ELC
      QTUB=ATUBO*.1714E-8*EMM*(TAVW**4-TS**4)
```

Figure E-1. Optimization Program Listing (cont'd)

```
      QFIN=QCOND-QTUB
      DCONEO=DCONE-2.*ELC*SIN(ACONE)
      ATOT=3.14*(DCONEO+DCONE)/2.*ELC
      AFIN=ATOT-TUBN*(DI+DO)/2./12.*ELC
      IF(AFIN.LT.0.0) GO TO 70
      EFFBAR=QFIN/(EMM*.1714E-8*(TAVW**4-TS**4)*AFIN)
      IF(EFFBAR.GT.0.99) GO TO 80
      DMEAN=DCONE-2.*ELC/2.*SIN(ACONE)
      CIRCF=3.14*DMEAN-TUBN*(DI+DO)/2./12.
      FINLT=CIRCF/TUBN/2.
      CALL D1DEG1(EFFBAR,A5,XLMBA)
      TFIN=EMM*.1714E-8*TAVW**3*FINLT**2/(XLMBA*CONF)*12.
      IF(TFIN.LT.0.001) GO TO 91
      IF(TFIN.GT.0.1) GO TO 92
      QSC=WD*CL*(TO-TOT)*60.
      IF(QSC.LT.0.0) GO TO 90
      COUNT2=0.0
  422 COUNT=0.0
   96 QCAC=0.0
      X3=-.05
      PIA=PI-DPIH-RHOV*VIN*VIN/9260.
      X8=XI
      X9=XI/10.
      DO 93 I=1,10
      X3=X3+.1
      X10=X8-X9
      D=DI-X3*(DI-DO)
      FINLT=(3.14*(DCONE-2.*ELC*X3*SIN(ACONE))-TUBN*D/12.)/TUBN/2.
      PIA=PIA-ELC/10.*DPOL(I) +RHOV*VIN*VIN/(XI**2*32.2*144.)*(X8**2-X10
     C**2)
      X8=X10
      IF(CODF.LT.1.5) CALL D1DEG1(PIA,A2,TO)
      IF(CODF.GT.2.5) CALL D1DEG1(PIA,A4,TO)
      IF(CODF.GT.1.5.AND.CODF.LT.2.5) CALL D1DEG1(PIA,A3,TO)
      TW=TO-DLT
      XLMBA=EMM*.1714E-8*TW**3*FINLT**2*12./(CONF*TFIN)
      CALL D1DEG1(XLMBA,A6,EFFLOC)
      QCAC=QCAC+EFFLOC*EMM*.1714E-8*TUBN*FINLT/6.*ELC/10.*(TW**4-TS**4)*
     C12.
   93 CONTINUE
      COUNT=COUNT+1.
      IF(COUNT.LT.1.5) GO TO 94
      IF(COUNT.LT.2.5) GO TO 97
      IF(COUNT.LT.3.5) GO TO 95
      GO TO 95
   94 IF(ABS((QFIN -QCAC)/QFIN ).LT..01) GO TO 95
      TFI=TFIN
      TFIN=TFIN*(1.+(QFIN -QCAC)/QFIN)
      IF(TFIN.LT.0.0) TFIN=TFI/2.
      TF2=TFIN
      QCONT1=QCAC
      GO TO 96
   97 IF(ABS((QFIN -QCAC)/QFIN ).LT..01) GO TO 95
      TFIN=TFIN-(QCAC-QFIN )*(TFIN-TFI)/(QCAC-QCONT1)
```

Figure E-1. Optimization Program Listing (cont'd)

```
      IF(TFIN.LT.0.0) TFIN=TF2/2.
      QCONT1=QCAC
      TF3=TFIN
      GO TO 96
   95 CONTINUE
      IF(ABS((QFIN-QCAC)/QFIN).LT..01) GO TO 421
420   COUNT2=COUNT2+1.
      IF(COUNT2.GT.6.5) GO TO 421
      TF4=TFIN
      TFIN=TFIN-(QCAC-QFIN )*(TFIN-TF2)/(QCAC-QCONT1)
      IF(TFIN.LT.0.0) TFIN=TF3/2.
      TF3=TFIN
      QCONT1=QCAC
      TF2=TF4
      GO TO 96
421   CONTINUE
      APRXL=ELC*QSC/QCOND
      REL=WD/(60.*TUBN*RHOL*3.14*DO**2)*4.*12.*DO*RHOL/VISL
      PR=CL*VISL/CONL*3600.
      HLIQ=.625*(REL*PR)**.4
      HLIQ=HLIQ*CONL*12./DO
      TW=((TO+TOT)/2.)-QSC*12./(3.14*APRXL*DO*HLIQ*TUBN)
      ATUB=APRXL*DO/12.*TUBN
      DCONO=DCONEO-2.*APRXL*SIN(ACONE)
      ATOSC=3.14*(DCONO+DCONEO)/2.*APRXL
      FINLT=(3.14*(DCONO+DCONEO)/2.-DO*TUBN/12.)*.5/TUBN
      XLMBA=EMM*.1714E-8*TW**3*FINLT**2*12./(CONF*TFIN)
      CALL DIDFG1(XLMBA,A6,EFFBAT)
      QACT=EMM*.1714E-8*TW**4*(ATUB+(ATOSC-ATUB)*EFFBAT)
      SCL=APRXL*QSC/QACT
      WTLQ=TUBN*3.14*DO**2/576.*SCL*RHOL
      AP=TUBN*3.14*(SCL*DO/12.+ELC*(DI+DO)/24.)/(3.281*3.281)
      TAU=835.
      TT0=3.68E-3*(TAU*AP/A10)**.273
      BT1=3.68E-3*(TAU*AP/A11)**.273
      TT2=3.68E-3*(TAU*AP/A12)**.273
      TT3=3.68E-3*(TAU*AP/A13)**.273
      TT4=3.68E-3*(TAU*AP/A14)**.273
      TT5=3.68E-3*(TAU*AP/A15)**.273
      TT6=3.683E-3*(1825.*AP/.001)**.273
      TT7=3.683E-3*(1825.*AP/.048)**.273
      TT8=3.683E-3*(1825.*AP/.195)**.273
      TT9=3.683E-3*(1825.*AP/.430)**.273
      TTW=3.683E-3*(1825.*AP/.700)**.273
      TTX=3.683E-3*(1825.*AP/1.12)**.273
      TOTL=SCL+ELC
      DCOE=DCONE-2.*TOTL*SIN(ACONE)
      ATOTAL=3.14*(DCONE+DCOE)/2.*TOTL
      AFIN=ATOTAL-ELC*(DO+DI)/24.*TUBN-SCL*DO*TUBN/12.
      WFIN=AFIN*TFIN/12.*RHF
      T1=TT0
      IF(T1.LT.TMINM) T1=TMINM
      TT0=T1
      T2=T1/4.
```

Figure E-1. Optimization Program Listing (cont'd)

```
IF(T2.LT.TMINM) T2=TMINM
CA0=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=BT1
IF(T1.LT.TMINM) T1=TMINM
BT1=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA1=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT2
IF(T1.LT.TMINM) T1=TMINM
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA2=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT3
IF(T1.LT.TMINM) T1=TMINM
TT3=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA3=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT4
IF(T1.LT.TMINM) T1=TMINM
TT4=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA4=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT5
IF(T1.LT.TMINM) T1=TMINM
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
TT5=T1
CA5=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT6
IF(T1.LT.TMINM) T1=TMINM
TT6=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA6=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT7
IF(T1.LT.TMINM) T1=TMINM
TT7=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA7=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT8
IF(T1.LT.TMINM) T1=TMINM
TT8=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
CA8=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
T1=TT9
IF(T1.LT.TMINM) T1=TMINM
TT9=T1
T2=T1/4.
IF(T2.LT.TMINM) T2=TMINM
```

Figure E-1. Optimization Program Listing (cont'd)

```
      CA9=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
      T1=TTW
      IF(T1.LT.TMINM) T1=TMINM
      TTW=T1
      T2=T1/4.
      IF(T2.LT.TMINM) T2=TMINM
      CAW=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
      T1=TTX
      IF(T1.LT.TMINM) T1=TMINM
      TTX=T1
      T2=T1/4
      IF(T2.LT.TMINM) T2=TMINM
      CAX=(DI+2.*T2)*(DI/2.+T1)+3.14*((DI+2.*T2)**2-DI**2*2.)/8.
      WT0=CA0*TOTL*RHT/144.*TUBN
      WT1=CA1*WT0/CA0
      WT2=CA2*WT0/CA0
      WT3=CA3*WT0/CA0
      WT4=CA4*WT0/CA0
      WT5=CA5*WT0/CA0
      WT6=CA6*WT0/CA0
      WT7=CA7*WT0/CA0
      WT8=CA8*WT0/CA0
      WT9=CA9*WT0/CA0
      WTW=CAW*WT0/CA0
      WTX=CAX*WT0/CA0
      QTOTL=QSC+QCOND
      WILH=.05*3.14/12.*DINH/12.*3.14*DCONE*RHT
      W0=WT0+WILH+WFIN    +WTLQ
      W1=WT1+WILH+WFIN    +WTLQ
      W2=WT2+WILH+WFIN    +WTLQ
      W3=WT3+WILH+WFIN    +WTLQ
      W4=WT4+WILH+WFIN+WTLQ
      W5=WT5+WILH+WFIN+WTLQ
      W6=W5-WT5+WT6
      W7=W5-WT5+WT7
      W8=W5-WT5+WT8
      W9=W5-WT5+WT9
      WW=W5-WT5+WTW
      WX=W5-WT5+WTX
      WRITE(6,100) DI,DO,TUBN,TFIN
      WRITE(6,101) EFFBAR,QTOTL,QCOND,QSC
      WRITE(6,102) TOTL,ELC,SCL,ATOTAL
109   FORMAT(10H       W4= ,F10.1,10H       W5= ,F10.1)
      WRITE(6,104) WFIN,TO,DPIH,WILH
      WRITE(6,112)
112   FORMAT(34H 20000 HR, 2 NINES MISSION WEIGHTS)
      WRITE(6,107) TT0,BT1,TT2,TT3
 107  FORMAT(29H TUBE WALL THICKNES N=0,1,2,3,4F12.3)
      WRITE(6,111) TT4,TT5
111   FORMAT(25H                    N=4,5,2F12.3)
      WRITE(6,103) WT0,WT1,WT2,WT3
      WRITE(6,424) WT4,WT5
424   FORMAT(10H  TUBW4 = ,F10.1,10H  TUBW5 = ,F10.1)
      WRITE(6,106) W0,W1,W2,W3
```

Figure E-1. Optimization Program Listing (cont'd)

```
      WRITE(6,109) W4,W5
      WRITE(6,423)
423   FORMAT(23H 5 YEAR 3 NINES WEIGHTS)
      WRITE(6,107) TT6,TT7,TT8,TT9,
      WRITE(6,111) TTW,TTX
      WRITE(6,103) WT6,WT7,WT8,WT9
      WRITE(6,424) WTW,WTX
      WRITE(6,106) W6,W7,W8,W9
      WRITE(6,109) WW,WX
      WRITE(6,108) QCAC,QFIN
      WRITE(6,419) WTLO
419   FORMAT(10H  WLO =    ,F10.1)
108   FORMAT(20H  ACTUAL COND FIN Q ,F14.1,15HREQUIRED FIN Q ,F14.1)
106   FORMAT(10H     WO  = ,F10.1,10H     W1 = ,F10.1,10H     W2  = ,F10.1
     C,10H    W3 =   ,F10.1)
100   FORMAT(10H0     DI = ,F10.3,10H     DO = ,F10.3,10H  NTUBES= ,F10.0
     C,10H  TFIN =   ,F10.4)
101   FORMAT(10H  FEFF  = ,F10.4,10H  QTOTL'= ,F10.0,10H  QCOND = ,F10.0
     C,10H  QSCLR=   ,F10.0)
102   FORMAT(10H  LTOT  = ,F10.3,10H  LCOND = ,F10.3,10H  LSCLR = ,F10.3
     C,10H  ATOT =   ,F10.3)
103   FORMAT(10H  TUBWO = ,F10.1,10H  TUBWI = ,F10.1,10H  TUBW2 = ,F10.1
     C,10H TUBW3 =   ,F10.1)
104   FORMAT(10H  FINW  = ,F10.1,10H TINSUB = ,F10.1,10H  DPIH  = ,F10.4
     C,10H  WTLH =   ,F10.1)
200   CONTINUE
      XT=XT+1.0
      IF(XT.GT.TN) GO TO 210
      TUBN=TUBN+DNT
      GO TO 220
210   XD=XD+1.
      IF(XD.GT.DN) GO TO 230
      DI=DI+DDII
      DO=DI
      DO=DO
      GO TO 240
 91   ERROR=7.
      GO TO 250
 92   ERROR=8.
      GO TO 250
800   ERROR=6.
      GO TO 250
 80   ERROR=1.
      GO TO 250
 30   ERROR=2.
      GO TO 250
 10   ERROR=3.
      GO TO 250
 70   ERROR=4.
      GO TO 250
 90   ERROR=5.
250   WRITE(6,260) DI, DO,TUBN,ERROR
260   FORMAT(10H0     DI = ,F10.3,10H     DO = ,F10.3,10H  NTUBES= ,F10.0
     C,10H  ERROR = ,F10.1)
```

Figure E-1. Optimization Program Listing (cont'd)

```
GO TO 200
END
```

Figure E-1. Optimization Program Listing (cont'd)

```
****  CONDENSING RADIATOR SIZING PROGRAM ****

 RHOL         HCONDS         KLIQ        RHOTUB        RHFI          RHFO
  42.8        10000.0        19.5         530.0        500.0         500.0
  TIN          THKR          KFINI        KFINO        TOUT          FLOW
1660.0           .4          198.0         13.7       1560.0          98.1
   R           GAMMA          HFG          TSNK        DCONE         LMAX
  39.5          1.6          823.0        565.0          7.3          23.1
  DP             PI          SURFT        ACONE         VISV          VISL
  .25          4.80          .0050       -.1745       .000012       .000098
  CPL           EMIS          CONF
  .18           .900           1.0

   DI  =       1.000      DO  =       1.000  NTUBES=        120.  ERROR  =          1.0

   DI  =       1.000      DO  =       1.000  NTUBES=        122.  ERROR  =          1.0

   DI  =       1.000      DO  =       1.000  NTUBES=        124.  ERROR  =          8.0

   DI  =       1.000      DO  =       1.000  NTUBES=        126.  TFIN  =        .0465
 FEFF  =       .8928  QTOTL  =    4954710.  QCOND  =   4858225.  QSCLR=        96485.
 LTOT  =      15.090  LCOND  =      14.801  LSCLR  =        .290  ATOT  =      470.993
 FINW  =       659.0 TINSUB  =      1649.3  DPIH   =       .0130  WILH  =         70.3
20000 HR, 2 NINES MISSION WEIGHTS
TUBE WALL THICKNES N=0,1,2,3            .231          .110          .083          .069
                   N=4,5            .061         .056
 TUBWO =      3427.7  TUBW1  =      2000.5  TUBW2  =      1807.5 TUBW3  =       1712.3
 TUBW4 =      1658.3  TUBW5  =      1620.0
   WO  =      4165.6     W1  =      2738.4    W2   =      2545.4    W3  =       2450.1
   W4=        2396.1     W5=        2357.9
5 YEAR 3 NINES WEIGHTS
TUBE WALL THICKNES N=0,1,2,3            .537          .187          .127          .103
   -               N=4,5            .090         .079
 TUBWO =      7672.4  TUBW1  =      2871.4  TUBW2  =      2154.2 TUBW3  =       1946.0
 TUBW4 =      1656.6  TUBW5  =      1780.9
   WO  =      6410.3     W1  =      3609.2    W2   =      2892.0    W3  =       2683.8
   W4=        2594.4     W5=        2518.7
 ACTUAL COND FIN Q       3080564.4REQUIRED FIN Q       3090012.6
 WLQ =          8.5
```

Figure E-2. Radiator Optimization Program Output

5. AN EXAMPLE CASE

The sample problem assumes that a potassium stream at 1200°F, flowing at 1.635 lb/sec, is to be condensed with an allowable two-phase pressure drop of 0.25 psia.

Designs are to be generated for a 1.0 inch inside diameter tube with the number of tubes varying from 120 to 160. Table E-3 gives SINDA input for this problem. Table E-4 shows FORTRAN data input cards. The corresponding output is shown in Figure E-2 for the first four tube number combinations.

Table E-3. SINDA Input for Sample Case

```
BCD 9CONDENSING RADIATOR TEST CASE
END
BCD 3CONSTANTS DATA
     1,.01,2, .15,3,.43,4,.83,5,1.28,6,1.8
END
BCD 3ARRAY DATA
     1,.01,1.35,.02,1.45,.04,1.6,.08,1.77,.1,1.85,.2,2.2,.4,2.8
     .6,3.3,.8,3.7,1.,4.,END $ LOCKHART-MARTINELLI CORR
     2,.15,1260.,.35,1340.,.76,1420.,2.,1540.,4.8,1660.
     10.,1720.,END$ K T VS P
     3,.025,1300.,.07,1480,.15,1460.,.3,1520.,.7,1620.
     1.,1660.,2.,1760.,END $SODIUM T VS P
     4,.6,1260.,1.5,1360.,3.,1460.,6.,1560.,10.1,1660.
     10.8,1760.,END $ CESIUM T VS P
     5,.06,100.,.1,45.,.2,10.,.25,6.,.3,4.,.35,2.9,.4,2.4
     .45,1.6,.5,1.2,.55,.93,.6,.7,.65,.55,.7,.41,.75,.3
     .80,.215,.85,.145,.9,.088,.95,.04,1.0,0.0,END $ FIN EFF
     6,0.0,1.0,.04,.95,.088,.9,.145,.85,.215,.80,.3,.75
     .41,.7,.55,.65,.7,.6,.93,.55,1.2,.5,1.6,.45,2.4,.4
     2.9,.35,4.,.3,6.,.25,10.,.2,45.,.1,100.,.06,END
END
BCD 3EXECUTION
     CONRAD(A1,A2,A3,A4,A5,A6)
END
BCD 3VARIABLES 1
END
BCD 3VARIABLES 2
END
BCD 3OUTPUT CALLS
END
BCD 3END OF DATA
```

Table E-4. FORTRAN Data Input Cards for Sample Problem

Col. 1 →10	Col. 11→20	Col. 21→30	Col. 31→40	Col. 41→50	Col. 51→60	Col. 61→70	Col. 71→80
42.8	19.5	1.E4	.25	4.8	1660.	530.	500.
198.	.005	-.1745	1560.	98.15	39.5	1.63	
1.23E-5	823.	.1835	7.32	.9	565.	23.1	1.
9.844E-5	500.	13.7	.4				
1.	0.0	1.	120.	2.	20.		

APPENDIX F

RADIATOR PERFORMANCE PROGRAM

The radiator performance program is described in this Appendix. Section 1, Program Abstract, gives a statement of the program's purpose and the calculation sequence. Section 2, Input Data List, defines the items required in the input data block. Section 3, Output Data List, defines terms used in the output data block. Section 4, Program Listing, gives a print-out of the program card deck. Section 5, An Example Case, gives the input and output data for an example case.

1. PROGRAM ABSTRACT

This program was written to calculate performance of a given condensing radiator geometry. Radiator inlet conditions are input and the program calculates outlet conditions. This program, in contrast to the optimization program, is included as an integral part of the overall systems analysis model as a subroutine, RADP. The performance subroutine, RADP, has ten arguments and requires four data cards for each case.

2. INPUT DATA LIST

The subroutine, RADP, requires incoming arguments provided by the main performance program and a data card block which is read at the start of the program. These inputs are defined in the following table.

Table F-1. LIST OF SYMBOLS USED IN DATA BLOCK

SYMBOL	DESCRIPTION	UNITS	LIMIT
Incoming Arguments			
TI	Inlet temperature	°R	
PI	Inlet pressure	psia	
WD	Flow rate	lb/min	
XI	Inlet quality		
Y	Flag used in systems program		
	y < 1.5 no data cards are read		
	y > 1.5 data cards are read		
III	Print flag		
	III < 1 write detailed output		
	III > 1 simplified output		
CARD 1 (8E10.5)			
RHOL	Liquid density	lb/ft^3	
CONL	Liquid K	Btu/hr-ft-°R	
RHF	Fin density	lb/ft^3	
RHT	Tube density	lb/ft^3	
CONF	Conductivity inner fin material	Btu/hr-ft-°R	

Table F-1. LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
SUFT	Surface tension	lb/ft	
ACONE	Cone angle	radians	
CARD 2 (7E10.5)			
R	Working fluid gas constant		
GAMMA	Specific heat ratio		
VISV	Vapor viscosity	lb/ft-sec	
CL	Liquid specific heat	Btu/lb-°F	
EMM	Fin emittance		
TS	Equivalent sink temperature	°R	
VISL	Liquid viscosity	lb/ft-sec	
CARD 3 (5E10.5)			
RHFO	Density of outer fin material	lb/ft^3	
CONFO	Conductivity of outer fin material	Btu/hr-ft-°R	
THKR	Outer to inner fin thickness ratio		
DCONE	Diameter of cone at inlet	ft	

Table F-1. LIST OF SYMBOLS USED IN DATA BLOCK (Cont'd)

SYMBOL	DESCRIPTION	UNITS	LIMIT
CODF	Fluid type code; 1.0 = potassium, 2.0 = sodium, 3.0 = cesium		
CARD 4 (4E10.5)			
TUBN	Number of tubes in radiator		
DI	Tube inside diameter	inches	
TFIN	Fin thickness	inches	
TLTH	Radiator tube length	ft	
ANGLE	Angle of cone taken up by a radiator segment	degrees	

3. OUTPUT DATA LIST

The subroutine, RADP, provides the main performance with output arguments, when called upon.

The program also gives identification of the radiator and fluid used as an input check. Outlet conditions are given for each segment of the condensing section and at the outlet of the radiator.

The output arguments and output parameters are defined below.

Output Arguments

T2	=	Outlet temperature (°R)
PO	=	Outlet pressure (psia)
CTUBLG	=	Condensing length (ft)
ERR	=	Error indicator: ERR = 1.0, valid solution ERR = 1.0, radiator cannot condense all vapor ERR = 2.0, condenser subcooler equations non-convergent

Output Parameters

TUBN	=	Number of tubes
DI	=	Tube inside diameter (inches)
TFIN	=	Fin thickness (inches)
TLTH	=	Total tube length (ft)
CODF	=	Fluid code
XOUT	=	Outlet quality from segment
QREJ	=	Heat rejected by segment (Btu/hr)
TW	=	Tube wall temperature of segment (°R)
EFF	=	Fin effectiveness of segment
TOUT	=	Outlet temperature from segment (°R)
TI	=	Radiator inlet temperature (°R)
TO	=	Radiator outlet temperature (°R)
PI	=	Radiator inlet pressure (psia)
PO	=	Radiator outlet pressure (psia)
WD	=	Flow rate (lb/min)

4. PROGRAM LISTING

A listing of the subroutine card deck is given below.

```
      SUBROUTINE RADR(TI,PI,WD,XI,Y,III,T2,PO,CTUBLG,ERR)
      DIMENSION A1(21),A2(13),A3(15),A4(13),A6(39)
      DATA (A1(I),I=1,21)/20,.01,1.35,.02,1.45,.04,1.6,.08,1.77,.1,1.85,
     C.2,2.2,.4,2.8,.6,3.3,.8,3.7,1.,4./
      DATA (A2(I),I=1,13)/12,.15,1260.,.35,1340.,.76,1420.,2.,1540.,
     C4.,1660.,10.,1720./
      DATA (A3(I),I=1,15)/14,.025,1330.,.07,1380.,.15,1460.,.3,1520.,
     C.7,1620.,1.,1660.,2.,1760./
      DATA (A4(I),I=1,13)/12,.6,1260.,1.5,1360.,3.,1460.,6.,1560.,
     C10.,1660.,30.,1760./
      DATA (A6(I),I=1,39)/38,0.0,1.,.04,.95,.08,.9,.145,.85,.215,.80,
     C.3,.75,.41,.7,.55,.65,.7,.6,.93,.55,1.2,.5,1.6,.45,2.4,.4,2.9,.35,
     C4.,.3,6.,.25,10.,.2,45.,.1,100.,.06/
      IF(Y.LT.1.5) GO TO 6
      READ(5,105) RHOL,CONL,CONH,RHF,RHT,CONF,SUFT,ACONL
 105  FORMAT(8E10.5)
      READ(5,108) R,GAMMA,VISV,     CL,EMM,TS,VISL
 108  FORMAT(7E10.5)
      READ(5,106) RHFO,CONFO,THKR,DCONE,CODF
 106  FORMAT(5E10.5)
   4  READ(5,107) TUBN,DI,TFIN,       TLTH,ANGLE
 107  FORMAT(5E10.5)
      ERR=0.
      RETURN
  6   PI=PI
      IF(III-1) 109,110,110
 109  WRITE(6,111)
      WRITE(6,112) TUBN,DI,TFIN,TLTH,CODF
  111 FORMAT(44H **CONDENSING RADIATOR PERFORMANCE PROGRAM**)
  112 FORMAT(6H0  TUBN=,F5.0,5H  DI=,F4.2,7H  TFIN=,F5.4,7H  TLTH=,F5.2,
     C7H  CODF=,F4.1)
      WRITE(6,113)
  113 FORMAT(39H0 CONDENSING SECTION INCREMENTAL VALUES)
      WRITE(6,114)
  114 FORMAT(10H0   XOUT  ,10H   QREJ  ,10H    TW   ,10H    EFF   ,10H
     C   TOUT   )
  110 CONTINUE
 500  FORMAT(F10.3,F10.0,F10.1,F10.3,F10.1)
C
      N=0
C
      T3=TI-459.6
      HFG=HVX(2,T3)
      RHOV=PI*144./(R*TI)
      VSONC=5.67*SQRT(GAMMA*R*TI)
      VIN=WD*XI*576./(60.*TUBN*RHOV*3.14*DI**2)
      IF(VIN.GT.VSONC) GO TO 50
      COUNT=0.0
      ERROR=0.0
      CONA=(THKR*CONFO+CONF)/(1.+THKR)
      DINH=.5*DI*SQRT(TUBN)
      REN=VIN*RHOV*DINH/(12.*VISV)
      DPIH=.316*3.14*DCONE*ANGLE/360.*RHOV*VIN**2/(4.*REN**.25*DINH*32.2
     C*24.)
      PI=PI-DPIH-RHOV*VIN**2/9260.
      DLTL=TLTH/10.
      X2=XI
      XL=-DLTL/2.
      QCOND=WD*60.*HFG*XI
      TEMP=QCOND/(3.14*DI*TUBN*TLTH*CONH)*12.
 100  XL=XL+DLTL
      FINLT=(3.14*(DCONE-2.*XL*SIN(ACONE))*ANGLE/360.-TUBN*DI/12.)/(TUBN
     C*2.)
```

Figure F-1. Listing of Subroutine Deck

```
      IF(CODF.LT.1.5) CALL DIDEG1(PI,A2,T2)
      IF(CODF.GT.2.5) CALL DIDEG1(PI,A4,T2)
      IF(CODF.GT.1.5.AND.CODF.LT.2.5) CALL DIDEG1(PI,A3,T2)
      TW=T2-TDRP
      XLMBA=EMM*.1714E-8*TW**3*FINLT**2*12./(CONB*TFIN)
      CALL DIDEG1(XLMBA,A6,EFF)
      Q=EMM*.1714E-8*(TW**4-TS**4)*(EFF*TUBN*2.*FINLT+TUBN*DI/12.)*DLTL
      T3=T2-459.6
      HFG=HVX(2,T3)
      DLTX=Q/(HFG*WD)/60.
      X2=X2-DLTX
      IF(X2.LT.0.0) GO TO 150
C
      N=N+1
C
      IF((XL+.5*DLTL).GT.TLTH) GO TO 200
      V=VIN*X2/X1
      THETA=SUFT*32.2/(VISV*V)
      RE=V*DI/12.*RHOV/VISV
      IF(THETA.GT.57.23.AND.THETA.LT.327.59) CE=4.182/THETA**.9225
      IF(THETA.GE.327.59) CE=.02
      IF(THETA.LE.57.23) CE=.1
      IF(III-1)501,502,502
  501 CONTINUE
      WRITE(6,500) X2,Q,TW,EFF,T2
  502 CONTINUE
      A=(((1.-X2)/X2)**1.8*RHOV/RHOL*(VISL/VISV)**.2)**.5
      CALL DIDEG1(A,A1,B)
      DPE=DLTL*CE*2./(RE**.2)*RHOV*V**2/(DI*32.2*12.)*B**2
      DPM=RHOV*VIN**2/(32.2*144.)*((X2+DLTX)**2-X2**2)/A1**2
      PI=PI+DPM-DPE
      GO TO 100
  150 CONDL=(X2+DLTX)/DLTX*DLTL
      SCOL=DLTL-CONDL
C
      CTUBLG=FLOAT(N)*DLTL+CONDL
C
      REL=WD/(60.*TUBN*RHOL*3.14*DI**2)*48.*DI*RHOL/VISL
      PR=CL*VISL/CONL*3600.
      HLIQ=.625*(REL*PR)**.4*CONL*12./DI
      AW=3.14*DI/12.*SCOL*TUBN
      ATUB=DI/12.*SCOL*TUBN
      AFIN=FINLT*TUBN*2.*SCOL
      TW=T2-50.
  350 Q1=EMM*.1714E-8*(TW**4-TS**4)*(ATUB+EFF*AFIN)
      TW2=T2-Q1*(2.+HLIQ*AW/(WD*60.*CL))/(2.*HLIQ*AW)
      IF(ABS(TW2-TW).LT.1.) GO TO 250
      COUNT=COUNT+1.
      IF(COUNT.GT.25.) GO TO 300
      TW=TW+.25*(TW2-TW)
      GO TO 350
  250 T2=T2-Q1/(CL*WD*60.)
      IF((XL+.5*DLTL).GT.TLTH) GO TO 400
      COUNT=0.0
      XL=XL+DLTL
      AW=3.14*DI/12.*DLTL*TUBN
      FINLT=(3.14*(DCONE-2.*XL*SIN(ACONE))*ANGLE/360.-TUBN*DI/12.)/(TUBN
     C*2.)
      XLMBA=EMM*.1714E-8*TW**3*FINLT**2*12./(CONF*TFIN)
      CALL DIDEG1(XLMBA,A6,EFF)
```

Figure F-1. Listing of Subroutine Deck (cont'd)

```
      ATUB=DI/12.*DLTL*TUBN
      AFIN=FINLT*TUBN*2.*DLTL
      GO TO 350
  250 CONTINUE
      ERR=1.
      RETURN
  300 WRITE(6,2) TI,WD,TW,TW2
    2 FORMAT(26H  CONDENSER NON-CONVERGENT,4F10.1)
      RETURN
   50 CONTINUE
      ERR=2.
      RETURN
  400 CONTINUE
      IF(III-1)402,401,401
  402 WRITE(6,115)
  115 FORMAT(35H0   RADIATOR INLET OUTLET CONDITIONS)
      WRITE(6,3) TI,T2,PI,PI,WD
    3 FORMAT(6H    TI=,F10.1,6H    TO=,F10.1,6H    PI=,F10.3,6H    PO=,F10.3
     C,6H    WD=F10.2)
  401 CONTINUE
      PO=PI
      ERR=0.
      RETURN
      END
```

Figure F-1. Listing of Subroutine Deck (cont'd)

5. AN EXAMPLE CASE

The radiator used for an example case is one of twelve in a larger capillary pumped system in which potassium is the working fluid. Table F-2 shows the input data and Table F-3 shows the corresponding output.

Table F-2. Data Input for Sample Problem

Col. 1 →10	Col. 11→20	Col. 21→30	Col. 31→40	Col. 41→50	Col. 51→60	Col. 61→70	Col. 71→80
42.8	19.5	1.E4	530.	500.	198.	.005	-.1745
39.5	1.63	1.23E-5	.1835	.9	565.	9.84E-5	
500.	13.7	.4	7.32	1.			
11.	1.	.0102	16.92	30.			

INPUT ARGUMENTS. T_{in} = 1660 °R

P_{in} = 4.8 psia

WD = 8.1 lb/min

XI = 1.0

II

Table F-3. Radiator Performance Program Output

```
**CONDENSING RADIATOR PERFORMANCE PROGRAM**

 TUBN=  11.  PI=1.00  TFIN=.0102  TLTH= 16.92  CODF= 1.0

CONDENSING SECTION, INCREMENTAL VALUES

 XOUT        QREJ        TW          EFF      TOUT
   .911      35687.      1656.8       .858     1657.6
   .819      37266.      1654.0       .825     1654.8
   .723      38685.      1652.1       .796     1652.9
   .623      40017.      1650.9       .765     1651.7
   .521      41245.      1650.2       .736     1651.1
   .416      42365.      1649.9       .707     1650.7
   .308      43442.      1649.8       .681     1650.6
   .198      44364.      1649.9       .655     1650.7
   .086      45122.      1650.1       .628     1650.9

 RADIATOR INLET OUTLET CONDITIONS
 TI=    1660.0    TO=     1549.5   PI=      4.800    PO=     4.590   WD=       8.15
```

APPENDIX G

NOMENCLATURE

A	Area
A_v	A_4
A_p	Heat transfer area in a capillary pump
A_t	Inside surface area of the radiator tube
A_1	Cross sectional area of the liquid line
A_4	Cross sectional area of the vapor line
C_e	Correlating parameter which gives the importance of surface tension on two phase friction.
$\bar{c}$	Speed of sound in armor material
d	Diameter of meteoroid
D_t	Diameter of the radiator tubes
D_1	Diameter of the vapor line
D_4	Diameter of the liquid line
f	Friction factor
h_c	Condensing potassium heat transfer coefficient
h_1	Working fluid vapor enthalpy at T_1 (saturated).
h_4	Subcooled working fluid enthalpy at the inlet to the capillary pump.
K	The cosine of the effective wetting angle
k	Thermal conductivity of the working fluid liquid
k_w	Thermal conductivity of the metal wall separating the condensing potassium and the capillary pump working fluid.
k_f	Thermal conductivity of the capillary pump working fluid

L	Length
L_v	Length of the vapor line = L_{12}
L_t	Spacing between radiator tubes
ΔL	Length of the radiator
m	Meteoroid mass
Mcrit	Minimum mass meteoroid which will give a penetration
N	Flux density of meteoroids or a working fluid property group ($\rho\ \lambda\ \xi/\mu$) or the number of capillary pumped loops.
n	Total number of impacts
Nu	Nusselt number (hD/k)
N_t	Number of radiator tubes
P	Pressure
Pe	Peclet number (Re)(Pr)
$P(N^1)$	Probability of N^1 or less punctures
P_1	Saturation pressure of the working fluid at T_1
P_4	Pressure at the inlet to the capillary pump surface tension
$\Delta P_{1\rightarrow 2}$	Pressure drop through the vapor line
$\Delta P_{3\rightarrow 4}$	Pressure drop through the liquid line
p	Probability of an impact resulting in a puncture
Q_P	Heat transfer rate in the capillary pump
Q_T	Heat transfer rate from the radiator
R	Effective radius of the wick
Re	Reynolds number = $\frac{\rho v D}{\mu}$
T_K	Condensing potassium temperature
T_o	Temperature of ultimate sink (space)

ΔT Temperature drop from the condensing potassium to the working fluid vapor

T_f Main can radiator fluid temperature

T_w Temperature of the radiator tube wall

T_{sk} Ultimate sink temperature = T_o

T_1 Working fluid vapor temperature leaving a capillary pump

T_2 Working fluid vapor temperature entering the radiator

T_3 Temperature of the liquid leaving the radiator

t Required armor thickness

U Overall heat transfer coefficient in a capillary pump

V Velocity

v Meteoroid velocity = 25 Km/sec (specified)

W The steady state mass flow rate in a capillary pumped loop

W_i The mass flow rate at point i in a capillary pumped loop

ΔX_w Metal wall thickness separating the condensing potassium and the capillary pump working fluid

ΔX_f Gap thickness between the metal wall and the screen wick

X_{in} Quality of working fluid into a radiator node

X_{out} Quality of working fluid out of a radiator node

a Damage thickness parameter = 1.75 (specified)

β Specified constant = 1.22

γ Material cratering coefficient = 2 (specified)

α Specified constant = $10^{-14.41}$ $gm^{1.22}/m^2 sec$

ε Emissivity of the radiator

ξ Surface tension of the working fluid

η Mean fin effectiveness

λ Latent heat of vaporization of the working fluid

μ_L Viscosity of liquid in the liquid line

μ_V Viscosity of vapor in the vapor line

μ Viscosity of the working fluid liquid

ρ Density

ρ_m Meteoroid density = 0.5 gm/cm^3 (specified)

ρ_t Armor material density

ρ_L Density of liquid in the liquid line

ρ_V Density of the vapor in the vapor line

ρ_1 Working fluid vapor density leaving the pump

ρ_4 Working fluid liquid density entering the pump

σ Stefan-Boltzmann's constant

τ Mission Time

ϕ^2 Martinelli-Nelson friction factor multiplier for two phase flow